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July 1959

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1958 Gas Turbine Progress Report

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INTRODUCTION

SIX YEARS have passed since the last progress report was written. These six years have brought the gas turbine to the forefront much sooner than many expected. Today gas-turbine power is being used in practically every type of application suitable to prime movers. In many cases it is used in an introductory manner; in other fields the gas turbine is well entrenched as these progress reports will indicate.

The authors of the reports are to be highly commended for producing factual data which have been selected from a tremendous amount of available information. The Gas Turbine Power Division is fortunate in having such highly qualified authors.

Six years ago this introduction gave a list of world-wide gas-turbine units. Today such a list would take many pages and would be inaccurate at best. Fortunately, many of the reports do give a rather comprehensive list. A map has been used to show the number of large gas turbines in stationary and mobile service other than aircraft. These units range mainly from 2000 hp to over 20,000 kw each. The four companies which have installed considerable heavy-duty equipment in the United States and Canada were selected. Those who are not familiar with the gas-turbine business will be surprised to note the large number in service and also the fact that these plants are so well scattered throughout the world.

The following is a very brief summary of the papers submitted. Complete details will be found in each of the papers.

Materials. Materials are one of the most important items in the gas-turbine field. No matter what methods are used to reduce the temperature of the gas turbine the best materials are preferable. Intensive studies have been made to reduce the cost of component parts by the use of alloys which are both cheaper and better. It is of interest that the report states, "some emphasis must be placed on the physical properties of metals at low ambient temperatures, especially when gas turbines are required to operate under adverse starting conditions" and a few seconds later shoot up to the abnormal high temperatures required.

Turbine Cooling. Turbine cooling has been very carefully studied during the past six years. Many tests have been made and certain results obtained. However, turbine cooling has not yet been used commercially except in a minor way. The desired breakthrough to materially increase turbine efficiency by using temperatures above 2000 F has not yet arrived commercially. Pametrada does have a turbine operating on 2200 F which has liquid cooling of the blades and apparently is operating very satisfactorily. Even this is not yet ready for the commercial market. Today it is an excellent research tool. The next six years should show a marked advance in turbine cooling and possibly turbine-inlet temperatures up to 3000 F.

Fuels. Gas turbines have been operated very successfully on natural gas, jet fuel, and distillates. However, the big difficulty comes with residual fuels and coal. The report on fuels gives an excellent account of progress to date on the burning of residual oil, while the progress report on locomotives includes a more detailed account of the burning of coal. The railroad and locomotive industries have been most anxious to find a means of reducing the cost of fuel over that of the diesel locomotive. Once the burning of coal has become commercial for a gas-turbine plant it will obviously be used more in the industrial and central-station industry than in the railroad field.

Cycle Components. Dr. Martinuzzi has done a remarkable job in picking up the loose ends for this general report. In the paper on cycle components, he has referred to several types of heat exchangers; the rotary-disk type, the rotary-wheel type, a reciprocating type as well as the stationary-plate type. The smallest gas turbine, the 7-kw Turbo-Mite, as well as many unusual designs of medium and large units, are described. There is also discussion on the use of the differential gear in various types of gas turbines for improving acceleration of the unit, particularly when attached to a vehicle.

Compound Piston-and-Turbine Engines. This report would not be complete without the chapter on compound piston-and-turbine engines as the gas turbine has been used to a major extent with the reciprocating engine. In fact, the compound engine requires the gas turbine in order to obtain its best results. These two when properly matched do produce a most efficient power unit. It should be mentioned that practically all modern diesel engines, even of the two-cycle type, are being turbocharged. This means the mass production of small gas-turbine component parts. The inventor of the turbocharger system, Dr. Alfred J. Buchi, was in this country recently and was extremely pleased to note—even though his patent has expired—that his system of turbocharging is being so universally used here as well as abroad.

Aviation. Dr. Lancaster emphasizes the importance of aviation. He itemizes only a few units in sufficient detail to show that of these five types of jet gas turbines one-half billion horsepower has been produced. Six years ago the jet was considered the prominent power plant for military aircraft. Today, it is just starting to replace the reciprocating engine in commercial aircraft. The jet unit has become so thoroughly accepted in recent years that no more reciprocating engines will be used on new large planes, commercial or otherwise. In other words, the gas turbine has taken over the aviation field whether it be used as a jet, a prop, or some combination. Today aviation is the gas turbine's biggest business.

Rocket and Missiles. Few realize the importance of the gas turbine in this field or the possible magnitude in the future. Practically every large rocket or missile requires an auxiliary power plant and the gas turbine, in most cases, is the ideal form for producing this power. It can be readily seen that, even when space ships come, the gas turbine will produce the base load even though it may have nothing to do with propulsion power.

Automotive. Here is a field that the gas turbine has not yet broken through. In the past six years at least seven manufacturers have produced one or more gas-turbine-propelled automobiles. Not one has yet become commercial. The present auto-

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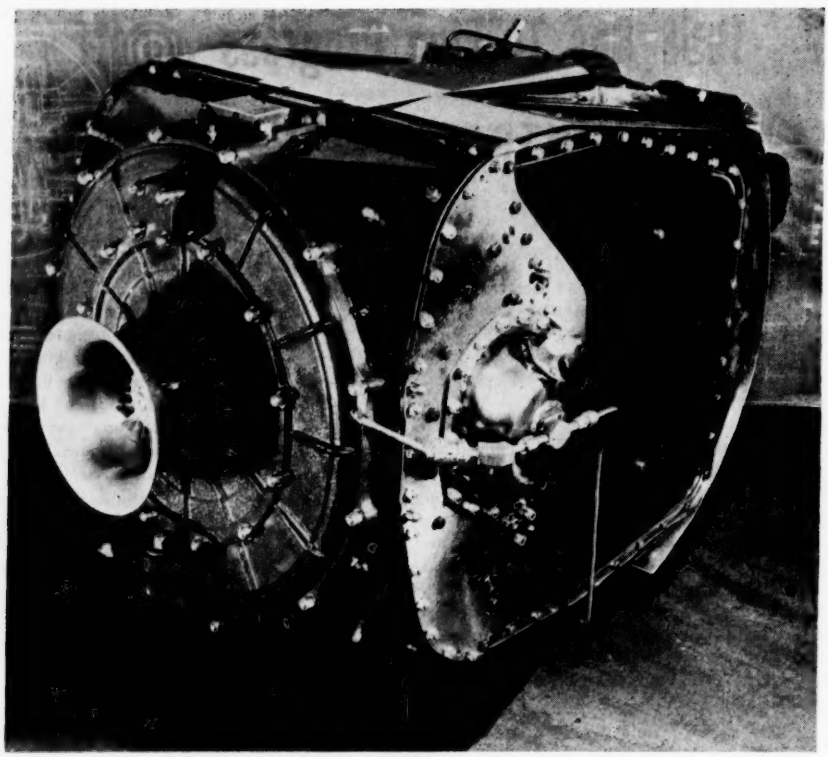


General Motors' latest gas-turbine power plant, the Whirlfire Model GT-305, now under development. This remarkable small unit with rotary heat exchanger is now operating on only 12.5 lb per hr of diesel fuel oil when idling. The full-load rating of 225 hp will make a weight ratio of less than 3 lb per hp. The fuel consumption expected at this rating is the unusually low figure of 0.55 lb per bhp per hr. Dimensions of package, 37 in. long, 27.8 in. high, 26 in. wide. Such a unit opens up a new era in the small-gas-turbine field—a unit that will fit practically every type of application—with economy.

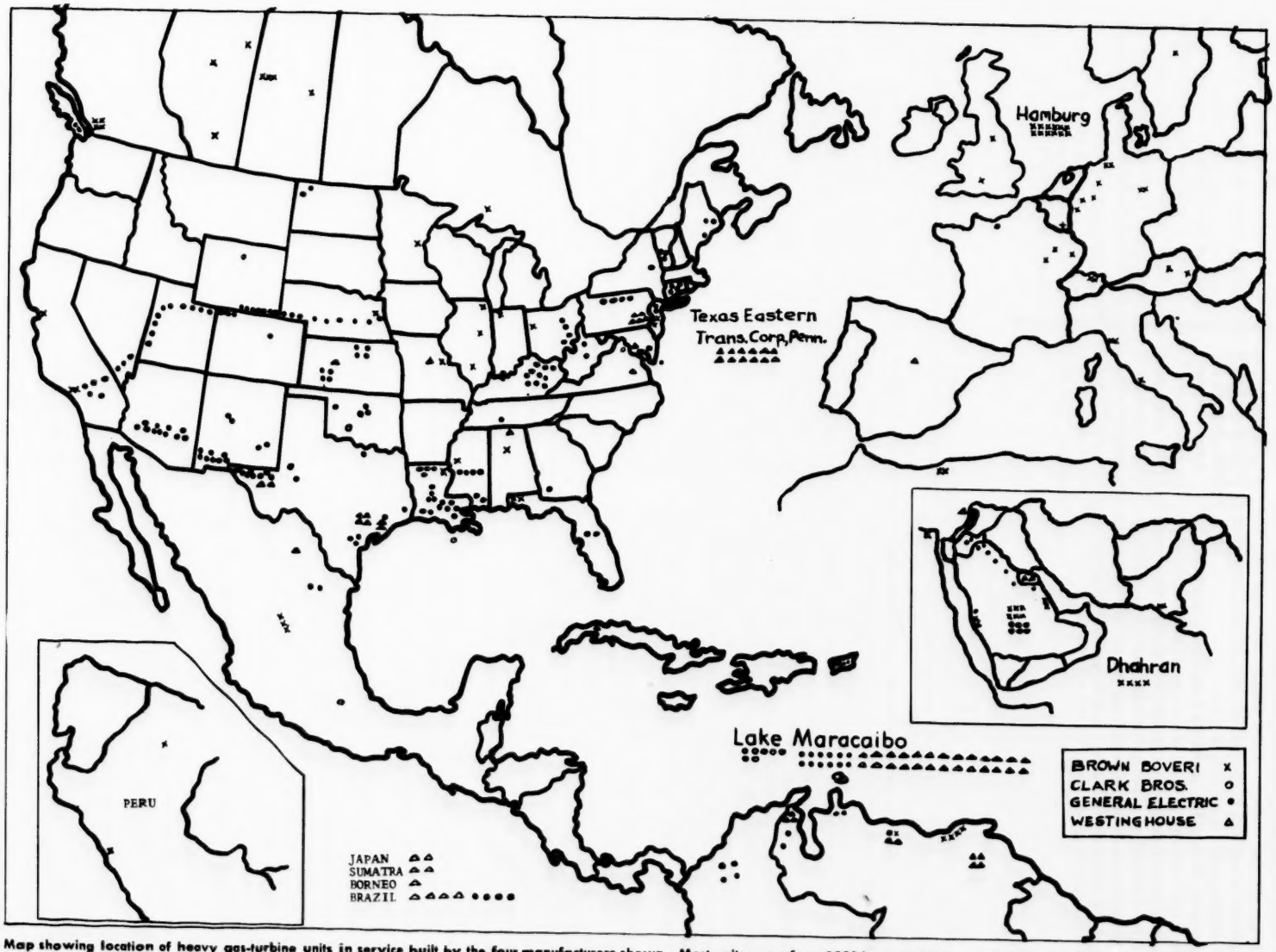
mobile is extremely flexible, powerful, and reliable—therefore difficult to improve. The gas-turbine automobile must be a better car. However, new gas-turbine power plants produced by the automotive industry are beginning to indicate that they can compete on both efficiency and weight-per-horsepower ratio. The major questions now are *performance and cost*. The gas-turbine auto still has a time lag in acceleration which is minor, but objectionable. The cost can be reduced mainly by quantity production.

Railroad. During the past six years the gas-turbine locomotive has become a success. Twenty-five 4500-hp units are now operating on the Union Pacific system and additional larger units have been ordered. Some of these new 8500-hp 3-unit locomotives should be in service by the time this paper is presented. The modern diesel locomotive is extremely versatile and economical, and it is difficult to produce a unit which can take its place. The gas-turbine locomotive has replaced it to a very limited extent. When the gas-turbine automobile has been definitely introduced, it is very likely that the gas-turbine locomotive will follow as both gas-turbine units are entering a highly competitive field. There are two types of coal-burning locomotives, one where the coal is burned in the combustor ahead of the power turbine, and the other where the coal is burned in the combustor after the power turbine but ahead of a heat exchanger which furnishes the hot gases to the turbine. The former is more efficient. The latter has the advantage of long blade life using a standard type of gas turbine as it does receive clean hot air. In fact, the air would be even cleaner than that used by turbines burning natural gas.

Marine. The marine paper is one of the longest in the series as so many types of turbines have actually been placed in service on the water. There has also been an extremely varied number of applications which range from high-speed light-weight boats, used particularly for short runs, to the ocean freighter. The former is a lightweight turbine without a heat exchanger. On the other hand ocean-going ships must be efficient and therefore require a large heat-exchanger unit. The many variations are covered in this paper.

Industrial and Stationary. Many of the industrial and stationary units are shown on the adjoining map previously referred to. Again we have a large assortment of shapes and sizes of turbines in this class of service. One of the most promising uses of the gas turbine in the future in this field is the supercharging of a steam-power-plant boiler. The gas turbine lends itself well to being a companion unit to other prime movers. It has assisted power output of the diesel so well that it may now help revolutionize and improve the steam-turbine plant. The gas turbine may also be used as a companion unit in the chemical industry. It has been used to a certain extent in the refinery field and may expand there. It is definitely expanding in the pumping of both gas and oil. One of its latest ventures is on the Trans-Arabian Pipe Line Company system where portable pumping plants of 5000 horsepower are utilized.

Nuclear Power. Little has been done in the nuclear gas-turbine field to date, but the possibilities of using gas turbines and high temperatures to produce a highly efficient nuclear power plant are considerable. General Dynamics has a contract from the Maritime Commission and the AEC to produce such a plant for



Map showing location of heavy gas-turbine units in service built by the four manufacturers shown. Most units range from 2000 hp to 20,000 kw. A few of the locations are process units in chemical or refinery plants. This includes the Brown Boveri units in the U. S.

eventual installation in a nuclear merchant ship. Possibly the only means of producing a nuclear power plant in a locomotive is by using the gas turbine. Even though little has been accomplished in this field in the past six years we can expect some very definite results in the next six years.

Conclusion. It would be interesting if the horsepower in the gas-turbine units referred to in all of the reports could be added up, but unfortunately the extreme variations in horsepower, applications, and individual types of turbines make this impossible. There was a day not so long ago when the gas-turbine industry was trying to become established and recognized. That era has come and gone in the past six years. Today the question is "How should we use the gas turbine," rather than, "should we use the gas turbine?" Today the gas turbine is probably finding its place steadily in more fields of application than any other two prime movers.

Acknowledgment. As editor of this extremely valuable 1958 progress report, I wish to express at this time my deep gratitude to the many authors and others whose contributions have materially assisted in its preparation. I am also particularly appreciative of the editorial work done on this report by Captain (Dr.) W. T. Sawyer, Office of Naval Research.

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MATERIALS'

Basic requirements for the selection of gas-turbine materials are defined. Alloy categories which meet the basic requirements for gas-turbine materials are discussed in detail. An evaluation of materials for critical gas-turbine components based on performance as well as limitations and future trends is presented. The dependence of future gas-turbine materials on the progress of gas-turbine metallurgy as well as gas-turbine design is strongly emphasized.

PRIOR to 1940 alloys with improved strength at elevated temperatures were largely developed for the petrochemical industries and steam power generation.

With the advent of World War II the development work on high-temperature alloys was greatly accelerated by the pressing need for satisfactory service and improved performance of gas turbines for military applications. The first noteworthy application was the improvement in reliability and performance of the turbosupercharger used in conjunction with the reciprocating aircraft engine. This was followed by the introduction and wide acceptance of the turbojet engine for military-aircraft propulsion and finally by some initial development of the marine gas turbine for ship propulsion.

It is very apparent, through thermodynamic considerations, that the simplest means of increasing gas-turbine cycle efficiency is through increased temperatures. Therefore, the rapid expansion of the gas-turbine industry has been accompanied by an equally extensive metallurgical program in the development of high-temperature alloys and refractory metals to permit higher turbine-inlet temperatures. Such development has reached a stage where gains in performance are not justified by the increased material costs, and methods of cooling component turbine parts are being given increased attention. Furthermore, although the major material problems associated with the design and operation of gas turbines are predominantly of a high-temperature nature, some emphasis must be placed on the physical properties at low ambient temperatures especially when gas turbines are required to operate under advance starting conditions.

Basic Requirements for Gas-Turbine Materials

In the selection of gas-turbine alloys it is essential to consider the intrinsic properties as well as the extrinsic properties and effect the best balance between these for operating conditions which are imposed through basic design.

Intrinsic Properties. These are the properties of an alloy which are inherent within the material because of composition limitations and are not materially affected by processing or heat-treatment and are as follows: (*a*) Density, (*b*) thermal expansion, (*c*) thermal conductivity, (*d*) modulus of elasticity, (*e*) oxidation resistance, (*f*) corrosion resistance, (*g*) damping capacity.

Extrinsic Properties. These refer to properties of an alloy which are considered somewhat variable and can vary in magnitude depending on melting, processing, and heat-treating procedures: (*a*) Tensile strength and ductility, (*b*) rupture strength and ductility, (*c*) creep strength, (*d*) fatigue strength, (*e*) impact strength, (*f*) structural stability, (*g*) hardness.

In addition to evaluating the two groups of physical properties and effecting the best balance in terms of either the short-time operation required by aircraft gas turbine or long-time operation required by land and marine gas turbines, it is essential that other considerations be factored when selecting an alloy for a given application. These are as follows: (*a*) Previous operational performance, (*b*) cost, (*c*) availability, (*d*) castability, (*e*) forgeability, (*f*) machinability, (*g*) weldability.

Although consideration and evaluation of the properties of an alloy represents a good first approximation in selection, much more emphasis and reliability must be placed on actual operation and past performance of a material under actual service conditions. This particularly applies to gas turbines where long-time operation is a primary requisite, and correlations of structural stability, thermal shock, fatigue strength, and similar properties based on short-time laboratory tests are very vague and unreliable.

The normal aircraft gas engine is representative of a turbine designed to maximum limits of stress, and in reality is designed for an operational service life of 1000 to 2000 hr before overhaul without much emphasis on economy in the selection of materials and construction. In contrast, the land gas turbine must be designed for 50,000 hr or more of operational service at a cost competitive with other industrial means of power generation and therefore presents a more challenging materials problem. In view of this, a greater emphasis will be placed on gas-turbine materials for long-time operation rather than for aircraft use.

Alloy Categories for Critical Gas-Turbine Components

Materials used in the design and construction of land gas turbines are extremely diversified, ranging from normal types of cast irons, cast steels, and low-carbon steels used for low-temperature and low-stressed parts to the critical superalloys used for applications involving both high temperature and high stress. However, the alloy steel used for critical gas-turbine parts may be classified into the four general groups commonly referred to as pearlitic or martensitic, ferritic, austenitic, and intermediate, which for purpose of clarification may be defined as follows:

1 Pearlitic or martensitic steels: (*a*) Those which undergo a transformation and are capable of heat-treatment, (*b*) those which may contain up to 15 per cent Cr with or without modifying

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elements: (1) Low alloy—containing up to 4 per cent alloying elements, usually in the form of Cr and Ni but modified with minor amounts of Mo, V, Si, and so forth; (2) intermediate alloy—containing 4.0 to 10 per cent Cr and usually modified with Mo, W, Al, and Si; (3) stainless—containing from 10 to 15 per cent Cr with modifications through Ni, Mo, W, and V additions.

2 Ferritic stainless steels: (*a*) Alloys which remain ferritic at all temperatures, (*b*) alloys which contain as low as 12 per cent Cr when modified with Al, but generally contain a minimum of 16 per cent Cr.

3 Austenitic stainless steels: (*a*) Alloys which remain austenitic at all temperatures, (*b*) alloys which generally contain a minimum of 16 per cent Cr and 8 per cent Ni—however, Mn and N can also be used as substitute elements for either a portion or total of the Ni content.

4 Intermediate stainless steels: (*a*) Those alloys which are austenitic at high temperatures and partially ferritic at low temperatures, (*b*) these are commonly known as the precipitation-hardening stainless steels and can be heat-treated by virtue of instability created in the austenite through alloy addition.

In terms of selection of materials for critical land-gas-turbine components, one is confronted with highly stressed parts operating from low ambient temperatures to 600 F on the compressor end, and from low ambient temperatures, as encountered in fast starting, to 1350 or 1400 F on the turbine end. In general the application of the various alloy classifications is as follows:

Low-Alloyed Steels. These steels are most frequently used in the heat-treated condition to meet the rigid physical requirements of highly stressed parts operating from low ambient temperatures to 1000 F. In general their application is limited to components which require a minimum of corrosion and oxidation resistance and the high-temperature strength requirements are relatively low. This classification may include such alloys as AISI 2340, 4140, and 4340 which exhibit good low-temperature toughness, the 1% Cr-1% Mo-0.25% V which shows relatively good low-temperature properties in combination with good high-temperature strength and oxidation resistance; and finally, the 3% Cr-0.5% Mo-1.25% Si which have a fair high-temperature strength but excel at the same time in oxidation resistance.

Intermediate-Alloy Steels. These steels are most frequently used in the normalized-and-tempered condition and find application in a temperature range of 900 to 1200 F where the high-temperature strength requirements are only moderate but where surface stability and resistance to oxidation and corrosion requirements are more stringent than can be obtained from the low-alloyed steels. Steels typical of this classification are the 4 to 6% Cr, 6 to 8% Cr all modified with Mo for increased high-temperature strength and in some cases with Si or Al for increased surface stability.

Moderately Alloyed Martensitic Stainless Steels. This classification of steels, commonly referred to as the hardenable stainless steels, has a relatively broad application in gas turbines because of their relatively good resistance to impositions of temperature, stress, corrosion, and oxidation encountered in gas-turbine-operation service.

The low-temperature toughness in the heat-treated condition, coupled with good over-all physical properties and resistance to normal-atmospheric as well as stress corrosion make these particular alloys very suitable for highly stressed compressor parts operating in a range of low ambient temperatures to 600 F. Typical alloys are represented in AISI 410, 420, and 431.

In addition, these alloys can be modified by the addition of such refractory elements as Mo, W, V, and Cb, which greatly improve their high-temperature strength characteristics, and thus provide a series of alloys with particularly good oxidation resistance for highly stressed turbine parts operating at 850 to 1100 F. Jessop H-46 and Crucible 422 are typical examples in this category of gas-turbine materials.

Moderately Alloyed Ferritic Stainless Steels. Steels of the ferritic type have never been too popular for high-temperature service because of their tendency to embrittlement and to grain growth at temperatures in excess of 1600 F. However, their low coefficients of expansion coupled with extremely good oxidation and corrosion resistance at elevated temperatures have dictated a limited use of these materials for extremely low-stressed gas-turbine parts. Typical applications are the AISI 430, 405, and 446 steels.

Moderately Alloyed Austenitic Steels. This classification of steels which includes many modifications of the commonly known 18% Cr-8% Ni stainless steels are exclusively used for gas-turbine parts operating in a temperature range of 1100 to 1500 F and at high or moderate stress levels dependent on the particular temperature involved. Most of these alloys can be readily fabricated by welding which offers an advantage over the martensitic and ferritic steels.

These particular alloys inherently show high coefficients of thermal expansion and low conductivity which are often considered unfavorable in turbine design. However, their high-temperature strength and excellent resistance to oxidation and corrosion render them invaluable for gas-turbine applications.

Generally these alloys may be divided into three groups, the oxidation and corrosion resistance being a function of alloy content:

Group 1—AISI 304, 316, 321, and 347
Group 2—AISI 309, 310, and 314
Group 3—35 Ni-15 Cr, 30 Ni-20 Cr, and 65 Ni-15 Cr

Highly Alloyed Superalloys. The superalloys are a complex series possessing the highest available strength at elevated temperatures. They have been continuously and extensively developed in the great efforts to increase the efficiency and capacity of the gas turbine through higher turbine-inlet temperatures. They have been applied to combustion components, turbine wheels, nozzle partitions, and turbine buckets.

In general the super alloys can be classified into four different groups depending on: (*a*) The nature and amount of the major alloying elements, (*b*) the relative amounts of minor alloying elements, and (*c*) the metallurgical mechanism through which high-temperature strength and stability are achieved. These four groups are generally represented by the so-called: (*a*) Iron-base, (*b*) nickel-base, (*c*) cobalt-base, and (*d*) chromium-base superalloys.

Iron-Base Alloys. These are in reality highly alloyed austenitic steels modified with Mo, W, and Ti, and represent the first attempts to produce gas-turbine materials with exceptional high-temperature strengths at 1200 F. However, it was found in development that warm working in a temperature range of 1200 to 1400 F was necessary to insure adequate yield strength at ordinary temperatures as well as to insure a high consistent rupture strength at 1200 F. Warm working is extremely critical in that it frequently results in an appreciable decrease in ductility at temperatures of 1300 to 1350 F, and may actually result in a loss in strength due to the precipitation of carbides accompanying strain aging in service. At higher operating temperatures solution treatment, with or without aging, does not appreciably improve the high-temperature strength. Typical iron-base alloys initially developed for gas turbines included Timken 16% Cr-25% Ni-6% Mo, 19-9 DL, 19-9 W-Mo, and Gamma Columbium.

Nickel-Base Alloys. Nickel-base alloys were initially developed for their corrosion resistance in the chemical industry and some of these alloys showed some promise for high-temperature application in the early development of gas turbines. In the rapid alloy-development period it was found that through modification of the normal Inconel (80% Ni-14% Cr-6% Fe) with Al and Ti a new series of superalloys was made possible. Later modifications included partial substitution of Co for Ni and the addition of such

refractory elements as Mo, W, and Cb. This series of alloys does not require warm working but attains high-temperature properties through solution treatment and subsequent precipitation hardening. At first, these alloys showed maximum strength with a single low-temperature age. However, subsequent tests and operational data indicated that a high intermediate age would considerably improve both the rupture ductility and structural stability with very little sacrifice in high-temperature strength. This treatment is being used on all new nickel-base alloys. The following nickel-base alloys have found wide acceptance in the gas-turbine industry: GMR 235, Inco 700, Inconel X, Nimonic Alloys, Waspalloy, Udimet 500, and M-252.

Cobalt-Base Alloys. This group refers to a complex series of Cr-Ni-Co-Fe alloys in which cobalt is either the only or one of the predominant alloying elements. These alloys are also modified by the addition of such refractory metals as Mo, W, and Cb. The high-temperature strength of these alloys is generally lower than the stronger nickel-base alloys. Since these alloys are of the solution-hardening type and strength is attained by a uniform distribution of refractory carbides in a stabilized austenitic matrix, these alloys have exceptional rupture ductility and have shown long-time structural stability far in excess of that indicated in the nickel-base alloys. Typical alloys that have been extensively used are X-40, Haynes Stellite 31, S-590, and S-816.

Chromium-Base Alloys. These alloys, still under development, are mainly confined to Cr-Fe-Mo and Cr-Fe-W systems modified with Ti and exhibit very high rupture strengths, but tend to be very brittle and show very poor rupture ductilities. However, it is felt that new methods of melting, deoxidation, and metal processing will greatly enhance their development.

Analysis of Critical Gas-Turbine Components

Each gas-turbine component must be capable of withstanding certain stresses at given temperatures for a minimum length of time while exposed to certain adverse corrosive and/or erosive environments. Materials for these components are accordingly selected by considering the possible operating conditions and comparing these with the available information on the properties of the materials. Special emphasis should be placed on those components which operate at high temperatures and high stresses or a combination of both (turbine end), as well as those operating at relatively low temperatures and high stresses or a combination of both (compressor end).

Turbine Wheels. Turbine wheels are one of the most critical of the turbine components from both a standpoint of operating requirements and material needs. They must have good high-temperature strength at the rim as well as low-temperature toughness in the center, particularly in designs which employ wheel cooling and/or require rapid starting in cold weather where thermal stresses of significant magnitude are superimposed. Although the best combination of high and low-temperature physical properties are found in the austenitic alloys, numerous factors, such as turbine-design size, manufacturing problems, cost, and the like, have dictated the use of ferritic alloys or a combination of austenitic and ferritic alloys (composite wheels) as well.

Composite Wheels. The composite wheel, first introduced by the General Electric Company in their original axial-flow jet engine and subsequently used for aircraft turbines, has also found considerable application in the small to intermediate-size commercial gas turbines having a limited number of turbine stages.

This type of wheel, consisting of an austenitic (Timken 16-25-6) rim welded to a pearlitic low-alloy-steel hub (Ni-Mo-V) represents a very economical construction and provides a good combination of high and low-temperature properties, particularly when the hub has been quenched and tempered. Thousands of such wheels have been used in aircraft gas turbines, and hundreds are currently giving good service in stationary gas turbines, some of which have accumulated as much as 40,000 hr of operation. In addition to proven reliability and initial economy, the composite wheel represents a design wherein overheated and damaged rims can be replaced on a finish-machined hub without much difficulty and at a minimum cost. A number of such wheels have had as many as four rim replacements.

The main shortcomings of this design are: (*a*) Size restrictions which result from procurement limitations of large 16-25-6 rims that have been properly hot-cold worked coupled with welding difficulties, and (*b*) the differential expansion of the rim relative to the hub.

At present the size of the composite wheel is governed by the rim size which is limited to 36 to 40 in. in diam at 800 to 1000 lb. The differential expansion which occurs between the rim and hub can result in the propagation of radial cracks from the roots of the dovetails, especially on machines which are subjected to excessive cyclic operation through numerous starts and stops. This, however, has been alleviated by radially slotting the rim to provide additional flexibility.

Austenitic Wheels. The use of solid austenitic wheels for land and marine gas turbines is also an outgrowth of the use of such materials by the aircraft-gas-turbine industry. The original use of a solid austenitic wheel in commercial units was restricted to very small multistage gas turbines. Because of the small size of the wheels involved, forged austenitic alloys of the Fe-base series such as 19-9 DL could be procured with some reliability and assurance of proper rim-and-bore properties.

Continued expansion and improvements in the melting and hot-working techniques of highly alloyed austenitic and superalloys has reached a stage where the application of such materials for turbine wheels to intermediate size multistage turbines and 2-stage small turbines has become a reality. In fact, such advancement has been so rapid, especially in consumable-electrode melting of these alloys, that the practical application of solid, highly alloyed austenitic alloys such as Discalloy and A-286 to larger size multistage gas turbines and medium-size 2-stage turbines is not too far away.

Although the austenitic alloys possess the best combination of high-temperature strength and low-temperature toughness—a primary requisite for turbine wheels—the special processing techniques required result in a very substantial increase in cost of the turbine-wheel component over that which might be accomplished with pearlitic or martensitic alloys ranging from low-alloy content to the stainless 12 per cent Cr type.

Ferritic Wheels. Although commonly referred to as ferritic alloys, materials of this classification which are used for turbine wheels are actually pearlitic or martensitic alloys by true definition and nomenclature. They can contain up to 15 per cent Cr with or without modifying elements and since they undergo a transformation, are capable of heat treatment.

The so-called ferritic wheels offer some advantage over the austenitic wheels providing temperatures are properly maintained at low levels. Although the ferritic alloys are somewhat deficient in high-temperature strength, they do exhibit moderate low-temperature toughness depending on the alloy content, melting and forging practice, and particularly the final heat-treatment. The advantage of the ferritic wheels over the austenitic wheels lies in their lower cost, greater consistency and reliability through known melting and forging practices, better machinability, lower coefficient of expansion, and better thermal conductivity.

Initial work in Europe on land and marine gas turbines had progressed along lines of using low austenitic alloys either in solid or welded construction; however, there has been a strong trend toward the use of so-called ferritic alloys ranging from as low as 1 per cent Cr to as high as 12 per cent Cr with minor alloy additions of W, Mo, and V. Likewise, limitations in construction of

composite wheels for medium and large 2-stage gas turbines and the procurement of reliable large austenitic forgings for the same machines as well as medium multistage machines has forced the American industry into the use of similar alloys.

It appeared that some modification of the 12 per cent Cr stainless steels would fulfill the needs for medium and large gas-turbine wheels. These series of alloys when modified with refractory elements, such as Mo, W, V, exhibit relatively high levels of smooth and notched rupture strength and show moderately good oxidation and corrosion resistance. The extensive use of Jessop H-46 and Rex 448 by the British for both aircraft and small stationary gas turbines indicated the potentialities of these alloys. In addition, the application of Lapelloy in the aircraft gas turbine in this country and production of large multistage steam and gas-turbine rotors of modified 12 per cent Cr steels in Germany definitely indicated a further trend in this direction.

In this country Crucible 422 was the first choice of material for medium and large gas-turbine wheels. This alloy, with a properly balanced composition to eliminate free ferrite, was first successfully used for 2nd-stage wheels in the small 2-stage turbines and subsequently, in the latter stages of small and medium multistage turbines. To date, as many as 16,000 to 18,000 hr of operation have been accumulated on the former-type machines.

In the case of the larger size wheels made from this particular alloy, it has been found that micro and macrosegregation presents very serious problems. These result from ingot solidification and incomplete breaking up during forging or lack of homogenization through heat-treatment. Excessive rejection rates dictate the use of an overspeed spin-test procedure to qualify each individual turbine wheel for service, since the degree and magnitude of segregation (representative of regions of low ductility) throughout the wheel cannot be detected by any established means of nondestructive testing. Furthermore, at the high strength levels employed to obtain maximum rupture properties at the rim, this alloy, in large sizes, exhibits inherently high impact transition temperatures (poor low-temperature toughness) at the bore, a condition not conducive to cold starting.

Because of these adverse experiences and industry's inability to produce a completely homogeneous forging of this alloy in large sizes at the present time even through the most advanced melting, casting, and forging techniques, other alloys of the intermediate class are currently being laboratory and service evaluated.

One of the most promising replacement alloys for Crucible 422 is the low-alloy 1% Cr-1% Mo-0.25% V composition. When quenched and tempered from the proper austenitizing temperature, it has excellent room-temperature strength and ductility, good smooth and notched rupture strength, and an impact transition temperature of 50 to 100 F or good low-temperature toughness. This alloy is much less prone to ingot segregation than the highly alloyed 12 per cent Cr alloys and can be produced to a higher degree of consistency and reliability through standard and established melting and forging techniques at an appreciably reduced cost as compared with Crucible 422. The alloy has 10 to 15 per cent lower 100,000-hr rupture strength when compared with Crucible 422 heat-treated to 145,000 to 150,000-psi tensile strength but compares favorably with Crucible 422 heat-treated to a 130,000 to 135,000-psi tensile strength. Efforts are being made to improve the rupture strength of this 1 per cent Cr alloy through heat treatment; which, if possible without embrittling tendencies, will preclude the necessity of design changes to accommodate it.

Other potential alloys for medium and large gas turbines which are less susceptible to segregation and should possess a good combination of high-temperature strength and low-temperature toughness are (*a*) 2% Cr-1% Mo-0.50% V, (*b*) 3% Cr-2% Mo-1% V, and (*c*) 4 to 6% Cr-1.5% Mo-0.75% V.

Future Trends. The future trend of materials for gas-turbine wheels is not clearly defined. One recourse is to extend the use of improved melting techniques such as the consumable-electrode process to all sizes that are required for gas-turbine wheels in order to provide austenitic and highly alloyed ferritic wheels which are homogeneous, free from gross segregation, and show a great degree of consistency and reliability through these specialized processing methods. This approach may be ruled out because of economical reasons. A second alternative is to promote the use of lower alloyed materials which can be produced satisfactorily by conventional methods to a high degree of reliability and at reduced cost. In cases where high-temperature strength is of primary concern, judicious wheel-space cooling or the use of long-shank turbine buckets might prove more effective as well as more economical.

Turbine Buckets. Turbine buckets (rotor vanes) experience the most severe combination of temperature and stress of any of the turbine parts. They must withstand very high centrifugal as well as superimposed vibratory stresses at elevated temperatures. In addition, they must withstand the blast of hot gases with entrained corroding products as well as exhibit good resistance to thermal shock when cyclic operation of the gas turbine is a primary requisite. In most cases turbine buckets operate at sufficiently high temperature and stress levels to require materials of the superalloy type. However, certain turbine designs, based on temperature and stress limitations, will permit the use of modified 12 per cent Cr alloys in the latter stages. Since these buckets are usually greater in length and more prone to the effects of vibratory stresses, some advantage may be gained from the greater damping capacity of the ferritic over the austenitic alloys. In the event that austenitic alloys are used throughout all stages, mechanical methods of damping, through the use of shrouds, tie wires, pins, or similar devices, are frequently employed.

S-590. One of the first superalloys used in appreciable quantities for stationary-gas-turbine buckets in this country was S-590. This alloy with a nominal composition of 20% Cr, 20% Co, 20% Ni, 4% Mo, 4% W, 4% Cb, 0.40% C, and Fe bal, is representative of a group whose high-temperature strength is achieved through solution hardening and subsequent stabilization aging. This alloy has exhibited good performance both for cyclic locomotive application with residual fuels as well as stationary units operating on distillate and natural gas.

S-816. This alloy, representative of the cobalt-base superalloys is, in reality, an improvement over the S-590 composition. This alloy with a nominal composition of 0.40% C, 20% Cr, 20% Ni, 4% W, 4% Mo, 44% Co, 4% Cb, and Fe bal, was extensively used in aircraft gas turbines prior to, and during, its adoption as a bucket material for industrial gas turbines. Considered the old reliable of the superalloys, with many years of proven service in diversified applications with a variety of fuels it is still extensively used on turbines where the inlet temperatures are in the vicinity of 1450 F.

Its greatest effectiveness has been shown on all oil-burning machines because of better resistance to oil-ash corrosion and thermal fatigue. Excellent structural stability and insensitivity to over-temperature deterioration, favor the long-time operation required of gas turbines. Destructive tests made on S-816 1st-stage buckets removed from an industrial turbine after approximately 38,000 hr failed to reveal any appreciable structural alteration and the rupture strength and ductility were well within the original acceptance standards.

Inconel X. This is an alloy representative of the nickel-base superalloys with a nominal composition corresponding to 0.04% C, 15% Cr, 73% Ni, 1% Cb, 2.5% Ti, and 0.60% Al. The high-temperature properties of this and other nickel-base alloys are obtained by a heat-treatment involving solution treating followed by precipitation hardening. Single treatment at 1300 F favors maximum strength or a double treatment consisting of heating at

1550 F prior to treatment at 1300 F sacrifices a little rupture strength in favor of rupture ductility. Inconel X has been used on small multistage turbines with moderately good results. Low rupture ductility which is aggravated by structural instability over long periods of operation is evident as an apparent drawback.

Nimonic 80A. This British version of a nickel-base superalloy was first introduced in this country on intermediate-size gas-burning turbines during the Korean war as a substitute for S-816 to conserve cobalt and columbium. Service experience showed that low rupture ductility coupled with structural instability as a result of a single treatment at 1300 F resulted in some bucket failures between 16,000 and 22,000 hr. A complete retreatment utilizing a double aging of 24 hr at 1550 F and 14 hr at 1300 F improved the rupture ductility and structural stability with little effect on the high-temperature strength. The nominal composition of Nimonic 80A is as follows: 0.10% C, 20.5% Cr, 75% Ni, 2.35% Ti, and 1.25% Al.

M-252. This alloy is also of the nickel-base superalloy class whose elevated temperature strength is attained through solution treatment followed by precipitation hardening. Some additional high-temperature strength is gained by addition of cobalt and molybdenum as indicated by a nominal composition of 0.15% C, 19% Cr, 55% Ni, 10% Mo, 10% Co, 2.5% Ti, 1.0% Al, and 2% Fe. Traces of boron and zirconium improve both the rupture strength and the rupture ductility of the alloy. When subjected to a double-aging treatment of 1550 F and 1400 F, it has very good notch ductility and fatigue strength. This alloy has been effectively used on gas-burning turbines operating at 1500 F inlet temperature because of rupture-strength limitations of S-816. Because of the high cost of vacuum-melting, it is only used where it offers a definite advantage over S-816.

Timken 16-25-6. Timken 16-25-6 alloy has been used with some limitations on 5000-hp industrial gas turbines. Although adequate in high-temperature rupture strength and oxidation resistance at the temperature involved, its adequate fatigue strength, presumably resulting from low damping capacity, resulted in premature dovetail failures. Replacement alloys of the modified 12 per cent Cr alloys were then used.

Crucible 422. With the exception of some early use of 12 per cent Cr steels modified with W and V, this alloy is now standard material for 2nd-stage buckets on 2-stage industrial gas turbines. It has good high-temperature strength in the range of 800 to 1000 F when initially heat-treated to a 140,000-psi tensile level. In addition, it has good oxidation resistance and favorable damping capacity and remains notch ductile at the high strength levels required for maximum rupture strength. Most of all, the alloy is very economical and presents no difficulties in processing forgings of the sizes involved for turbine buckets.

Future Trends. If the trend for higher inlet temperatures continues, more emphasis will probably be placed on higher strength nickel-base alloys, such as Udimet 500 and 600. Long-time creep-rupture tests as well as stability tests are now under way. Another approach under dual consideration is to reduce bucket stresses by weight reduction; namely, the use of hollow buckets. This would permit continued use of now-established and proved alloy compositions for turbine buckets especially for the higher-temperature stages. Although there has been appreciable mention of turbine-bucket cooling, actual activity along this line, especially applicable to industrial turbines, has been extremely slow.

Since Crucible 422 affords little margin for future increases in stress and temperature for latter-stage buckets, it appears that such increases will require the consideration of economical austenitic alloys such as A-286 with added provisions for mechanical damping.

Nozzle Partitions. Nozzle partitions, or stator vanes, although critical turbine components, do not possess very rigid material requirements by virtue of the relatively low stress levels at which they operate. A good partition material generally must meet the following requirements: (*a*) Good oxidation and/or corrosion resistance, (*b*) resistance to thermal fatigue, (*c*) good weldability, (*d*) thermal coefficients commensurate with dimensional stability, and (*e*) favorable structural stability.

Generally nozzle partitions are made from either the moderately alloyed martensitic or ferritic stainless steels or the moderately alloyed austenitic stainless steels ranging from Group 1 to Group 3. In applications of severe corrosion encountered in turbines burning residual fuel, almost every alloy is readily attacked by the fuel ash, and therefore treatment of the fuel oil is a necessity to minimize the attack. With reduced corrosion rates other problems such as thermal fatigue and stability become more evident.

Type 310 Stainless. This alloy of 25 Cr-20 Ni is a good combination of oxidation resistance, adequate high-temperature strength, weldability, and availability, and has been successfully used for 1st-stage partitions of 2-stage turbines and the early partitions of multistage turbines which use gaseous-type fuels. This same analysis failed miserably on units burning residual fuel apparently due to V_2O_5 attack as well as sulfidation aggravated by intermittent overheating effects.

Type 314 Stainless. Type 25 Cr-20 Ni alloy modified with silicon shows improved performance over type 310 in gas-burning industrial turbines. Although more susceptible to sigma embrittlement with time, the higher silicon content reduces the coefficient of thermal expansion and improves grain-boundary oxidation resistance so as to improve the resistance to thermal fatigue. Some degree of success for short-time operation has been obtained in oil-burning machines; however, rigid limitations on over-heating are necessary to prevent rapid attack and deterioration due to corrosion.

Type 446 Stainless. The straight chromium ferritic steels have always shown good resistance to corrosion in burning residual fuel. Type 446 stainless containing 28 per cent Cr is currently used on all residual-fuel-burning machines operating with high turbine-inlet temperatures. Low thermal expansion gives a remarkable resistance to thermal fatigue, but has the disadvantage of low creep resistance and is considered unweldable due to grain growth and accompanying embrittlement.

The weakness of this alloy in creep—evidenced by bowing of the trailing edges by the gas stream—has precipitated the search for other alloys that may be used with properly treated residual fuels.

X-40 (H. S. 31). The nominal composition of this cobalt-base alloy is 0.50% C, 24.50% Cr, 10.50% Ni, 7.5% W, 55% Co, and 1.0% Fe. Small burner oil-ash corrosion tests have indicated fair resistance to corrosion of treated residual fuels as compared to type 446 providing temperatures are maintained below 1550 F. In excess of this temperature, sulfidation from the presence of cobalt and nickel is very pronounced.

Good resistance to thermal fatigue, excellent creep-rupture strength and excellent weldability make it attractive as a partition material. Current evaluation of this material under service conditions is now being made.

Type 304 Stainless. Although modified 18-8 stainless steels such as type 321 and type 347 have frequently been used for 2nd and 3rd-stage nozzle partitions, no particular advantage of these alloys over type 304 is evident. The latter is entirely suitable for the application intended and is more readily procurable at reduced costs.

12 Per Cent Chromium Stainless. Partitions made from these grades of alloys have found considerable application in the latter stages of small and medium multistage industrial gas turbines. These alloys may be of the 403 type with a balanced analysis which favors weldability or the ferritic 405 type containing aluminum which is capable of being welded with added precaution.

Future Trends. For oil-burning machines the best approach for improved nozzle performance from a standpoint of corrosion appears to be proper and improved methods of oil treatment. However, concurrent with this, some gain may be made with alloys and designs aimed at minimizing thermal fatigue. Modifications of type 446 to improve corrosion resistance appear to be in order. From a design standpoint, the hollow partition seems to offer the best advantages.

For gas-burning machines alloys with better oxidation resistance, lower expansion, and better dimensional and structural stability are being investigated. These include Incoloy, Inco 804, 15 Cr-35 Ni, 15 Cr-65 Ni, and Inconel.

Compressor Rotors. Compressor rotors are generally built upon individual wheels and stub shafts properly stacked by welding, shrinking, or through-bolting to produce the final rotor. Since the compressor operates in a range from low ambient temperatures to a maximum of 600 F, the primary requisite of the wheel materials is uniformity in physical properties, soundness, and low-temperature toughness.

Generally speaking, all compressor wheels are made from standard low-alloy steel. The composition depends on the section size and corresponding hardenability of the alloy and the degree of low-temperature toughness which is required under the lowest operating ambient temperature. These analyses vary from AISI 4130 for the smallest wheels made from closed-die forgings to the armor-plate compositions containing appreciable amounts of nickel together with Cr, Mo, and V for large wheels and stub shafts made from flat open-die forgings. In any event, all compressor wheels are oil-quenched and tempered to obtain optimum physical properties commensurate with low-temperature toughness.

Where added corrosion resistance is deemed essential, the wheels are generally electroplated with cadmium and baked for removal of hydrogen.

Compressor Blades. The geometrical shapes of stator and rotor compressor blades, required to enhance their aerodynamic characteristics and performance, render them more susceptible to vibration and foreign-object damage than other turbine components. Consequently, they must be made from a material with relatively high levels of internal damping, fatigue strength, impact strength, ductility, and corrosion resistance. Type 403 stainless fits these requirements better than any other commercially available material. The reliability of this alloy has been proved through many years of satisfactory operation in steam-turbine buckets, and both aircraft and stationary-gas-turbine compressor blades.

This alloy is readily available at moderate cost, and lends itself to many diversified methods of fabrication. Consequently, compressor blades have been produced very economically by such methods as machining, forging, extruding, fabricating by welding, and/or brazing and coining.

Occasionally environments are encountered which dictate the use of an alloy with better corrosion resistance or improved abrasion resistance. Such applications, although limited in number, are very difficult because they generally require sacrificing one or more of the desirable properties of type 403 stainless.

One such application arises in the operation of gas-turbine locomotives. Limitations in the amount of air-cleaning or filtering equipment because of space requirements result in a dirt-entrained atmosphere which erodes the blades to a point where loss in compressor efficiency and turbine output requires replacement after approximately 15,000 hr.

Original erosion tests conducted on compressor-blade material at relatively high impingement angles seemed to indicate that hardness bore no relationship to erosion rates, and inorganic coatings of good resilience proved most noteworthy. Field tests on such coatings failed from inability to adhere to the metallic surfaces under conditions of actual operation.

It now appears that erosion of compressor blades can be retarded, if not completely eliminated, by one of the following: (*a*) Selective hardening of the tips of the rotor blades and root sections of the stator blades; (*b*) proper application of hard metallic coatings by electroplating or flame deposition; and (*c*) the use of alloys with a properly balanced microstructure—uniform carbides in a relatively ductile matrix. Various field tests are now under way with preliminary, but very assuring, results.

Where extremely corrosive atmospheres prevail, type 403 has its limitations depending on the corrosive environment and the loss in fatigue strength due to stress concentration set up by surface attack.

Future Trends. Potential alloys for compressor blading at higher strength levels with good resistance to corrosion fatigue and moderate resistance to corrosion appear to be in the modified 12 per cent Cr alloys such as Crucible 422. In the chemical industries and marine service where resistance to corrosion is of primary importance the more corrosion-resistant alloys capable of heat treatment appear most promising. These include the precipitation-hardening alloys such as 17-4 PH, 17-7 PH, AM 350, and AM 355.

Combustion-Chamber Liners. The primary requisites of combustion-chamber liner material are: (*a*) Good oxidation resistance, (*b*) good corrosion resistance to fuel ash, (*c*) dimensional stability as governed by expansion characteristics, and (*d*) fabricability at lowest possible costs.

The British gas-turbine industry has relied on such alloys as Inconel and Nimonic 75 for liner materials, while in this country more conservative alloys, such as type 309S and type 310S have been used more extensively than Inconel with a great degree of success.

The performance of gas-burning machines requires little, if any, servicing or replacement, and turbines operating upwards of 40,000 hr still have their original combustion liners. Oil-burning machines, on the other hand, are more susceptible to erratic combustion and, hence, have a higher rate of failure due to excessive distortion burnouts and general deterioration. Such failures are frequently so drastic as to defy the resistance of any known material and therefore represent a problem which must be solved through improved methods of fuel combustion.

Although the aircraft-gas-turbine industry is approaching the use of such superalloys as N-155 for liner materials, it is felt that alloy development for stationary gas turbines is not justified at the present time.

Summary

Stationary land and marine gas turbines have definitely established themselves in the petrochemical industries, the power-generation field, and as a prime mover for both locomotive and ship propulsion. The future material needs for stationary gas turbines will be greatly influenced by the long-time performance and service records of units now in operation coupled with the availability of reliable design data and processing data on newly developed alloys. Future gas-turbine materials will, in addition, depend upon the technological advances in gas-turbine metallurgy that can be effectively utilized in conjunction with progress made in gas-turbine engineering and design.

Active development of nonferrous alloys, cast irons, cast steels, or heat-treatable alloys of the low-alloy or intermediate-alloy grades does not appear justified at present. These materials, generally used at atmospheric conditions, have been used and will continue to be used in gas turbines based on properties and requirements that are also suitable for other applications.

New developments in the highly alloyed austenitic materials and superalloys, especially in the refractory metals, may be anticipated because of technological advances in melting and casting techniques. Greater emphasis will be placed on the heat-

treatment of such alloys to insure optimum high-temperature strength with good rupture ductility and long-time structural stability.

The development of cermets, ceramic coatings, and refractory metal coatings has reached a stage where application in short-life aircraft gas turbines is feasible. However, their brittleness and limited stability over long periods of operation render them unsuitable for stationary gas turbines at the present time.

In conclusion, the future gas-turbine materials will represent a compromise between progress in gas-turbine metallurgy and gas-turbine design. The use of proper design criteria with adequate margins of safety will invariably result in more efficient and reliable performance of alloys in gas turbines. On the other hand, there must be an active participation in alloy development and reduction in cost of critical gas-turbine components either through improvement in properties of cheaper alloys or through improved methods of processing. Both of these will greatly enhance the use of stationary gas turbines for diversified commercial application.

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TURBINE COOLING

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Turbine cooling, originally developed because of a shortage of heat-resistant alloys, is of interest for certain applications to permit operation of gas-turbine engines above uncooled temperatures. Specific power output would be increased, and in some cases specific fuel consumption decreased, depending on the engine type. Progress is reported on developments since 1952.

Turbine cooling has been of some interest for quite a number of years as a means of permitting increases in turbine-inlet temperature and as a means of permitting the use of less strategic materials in the gas turbine. Probably the earliest reference to cooling of gas-turbine blades was by Sir Charles Parsons in British patent number 6735 of 1884 [1].[1] Serious development of turbine-cooling methods started in Germany about 1935 with development of exhaust-gas turbosuperchargers for use in high-altitude airplanes. Means of cooling the blades to withstand the exhaust-gas temperatures of 1800 to 2200 F were developed at the beginning of World War II when the Germans started the development of turbojet engines, cooling was also used for the turbine blades even though the gas temperatures were below those for turbosuperchargers. Cooling was used because of the extreme shortage of heat-resistant alloys in Germany during the war. By the end of World War II all of the German turbosuperchargers and turbojet engines in operation, production, development, or proposed for the future, were equipped with cooled, hollow turbine blades [2].

In most other countries cooling has not played such a prominent part because better heat-resistant materials were available. Nevertheless, there is and has been an interest in operating gas-turbine engines at temperature levels in excess of those permissible with uncooled turbines in order to increase specific power output or to decrease specific fuel consumption, depending on the engine type. This paper will present a progress report on gas-turbine cooling, dealing principally with developments made available in the unclassified literature since the 1952 progress report by Hafer [3].

Effect of Turbine-Inlet Temperature on Engine Performance

Increasing the turbine-inlet temperature of a gas-turbine engine results in very significant increases in power output. A performance comparison of turboprop and afterburning and nonafterburning turbojet engines is shown in Fig. 1 for a range of turbine-inlet temperatures [4]. In order to compare all engines on the same basis, the shaft power of the turboprop engine was converted to thrust by assuming that a propeller thrust was added to that of the jet thrust. Since the comparison is on a relative rather than an absolute basis and is for a constant flight velocity, the results shown could also be interpreted as being based on specific power since power is a product of thrust and velocity. These performance curves were calculated by the methods of Reference [5]. The flight Mach number of 0.9 used for the comparison is a compromise value that could be feasible for both turbojet and turboprop engines. The performance shown is relative to that of a nonafterburning turbojet engine with a 1540-F turbine-inlet temperature. All engines were assumed to have a compressor-pressure ratio of 12 and compressor and turbine adiabatic efficiencies of 0.85. Afterburning temperature was assumed to be 3000 F. The effects of blade cooling are shown by the cooling effectiveness parameter ζ which is defined as

$$\zeta = \frac{T_{a,\,out} - T_{a,\,in}}{T_B - T_{a,\,in}}$$

where

$T_{a,\,in}$ = cooling-air inlet temperature at blade base
$T_{a,\,out}$ = cooling-air temperature at blade tip
T_B = average blade temperature

This parameter is a measure of how well the heat capacity of the cooling air is utilized, and in evaluating the effect on engine performance it was necessary to know the gas-to-blade heat-transfer coefficients which can be obtained from [6]. For a good air-cooled blade design the value of ζ approaches or exceeds a value of 1.0. A poor cooling design might have a value of $\zeta = 0.5$. The performance curves for an engine with uncooled blades would be somewhat better than those shown for $\zeta = 1.0$.

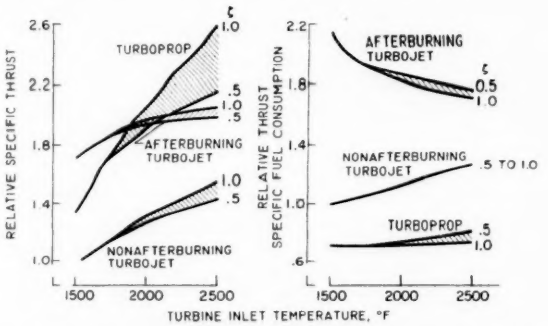

Fig. 1 Effect of turbine-inlet temperature on relative specific thrust and relative-thrust specific fuel consumption. Flight Mach number 0.9, altitude 40,000 ft (effects of air cooling included).

From Fig. 1 it can be seen that the thrust of turboprop and nonafterburning turbojet engines can be increased greatly by increasing turbine-inlet temperature. For the turboprop engine there is only a slight increase in specific fuel consumption. The specific fuel consumption for nonafterburning turbojet engines increases more with turbine-inlet temperature, but it is still more economical than using an afterburner. Afterburning to 3000 F results in the highest thrust by far for the three types of engines at low turbine-inlet temperatures, but the specific fuel consumption is very high. The use of higher turbine-inlet temperatures reduces the specific fuel consumption of afterburning engines and at the same time results in increased thrust.

[1] Numbers in brackets designate References at end of paper.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, June 3, 1958. Paper No. 58—A-46C.

From a performance point of view, therefore, there appear to be no disadvantages to increasing turbine-inlet temperatures for turboprop and afterburning turbojet engines. The cooling effectiveness of the turbine blades should be high, however, for the turboprop to get the greatest increases in power and at the same time not cause the specific fuel consumption to increase. For nonafterburning turbojet engines, Fig. 1 shows that increases in relative specific thrust are accompanied by increases in relative specific fuel consumption. The rate of increase in relative specific thrust, however, is about twice that of the increase in relative specific fuel consumption. Other studies have shown that for nonafterburning turbojet engines at supersonic flight speeds of about Mach 1.5 to 3, the higher thrusts per pound of air flow and the reduced engine weight per pound of thrust, as turbine-inlet temperature is increased, overcome the effect of increased specific fuel consumption. As a result, increased inlet temperatures are desirable for this flight application.

The comparisons made in Fig. 1 are of a general nature only and do not necessarily include the optimum condition for any of the power plants. For the turboprop or gas-turbine engines used for land or marine power plants, the performance (particularly fuel consumption) can usually be improved by increasing the compressor-pressure ratio as the inlet temperature is increased. For supersonic flight with turbojet engines it is usually desirable to have pressure ratios less than the value of 12 used in Fig. 1. The figure does, however, indicate the general trends that can be expected with increased inlet temperature, and it shows that for many applications increasing turbine-inlet temperature would be desirable.

Quite a number of both analytical and experimental studies [7] to [16] have been made to evaluate the effects of cooling on performance of various types of gas-turbine engines. In [16] effects of discharging cooling air from the tips of turbine blades are evaluated. It was found that the cooling-air discharge can exert a thrust on the blades and improve turbine efficiency.

Turbine Blade Materials

Since the development of the gas-turbine engine there has been a steady improvement in the high-strength properties of turbine blades as new alloys have been developed. Fig. 2 shows the 100-hr stress-rupture properties of typical alloys that were available in the years 1945, 1952, and 1958. Data were obtained from [17] through [22]. It will be noted that by use of these newer alloys it is possible to either: (*a*) Increase the turbine-inlet temperature for a given blade-stress level; (*b*) operate essentially at the same turbine-inlet temperature and stress level, but improve the reliability of the turbine through use of a greater margin of safety; or (*c*) operate turbines at higher stress levels which is often desirable in high-performance engines using new compressor developments such as the transonic compressor.

If, for the moment, we consider alloys other than molybdenum alloys it can be seen that from 1945 to 1958, for a stress level of 20,000 psi which might be considered typical at the critical section of a turbine blade, the maximum permissible increase in blade-metal temperature is about 200 F. Although this increased inlet temperature can improve the performance of gas-turbine engines, much greater performance improvements are possible by going to higher inlet temperatures as shown in Fig. 1. The use of molybdenum alloys has been suggested and the stress-rupture properties of a molybdenum alloy containing 0.45 per cent titanium are shown in Fig. 2. By use of such alloys for turbine blades, turbine-inlet temperatures in excess of 2000 F should be possible. The difficulty with using molybdenum, however, is that it oxidizes severely at elevated temperatures and some type of protective coating is required. Considerable research has and is being conducted on such coatings but, unless the coating is self-healing, even a pinhole through the coating, caused by a foreign object passing through the turbine, can destroy the blade through oxidation because the oxide is a gas and will pass out through any small defect in the coating. It appears, therefore, that further research to develop a self-healing coating for molybdenum is required before molybdenum turbine blades can be used effectively in long-life gas-turbine engines.

In order to utilize the desirable higher gas temperatures indicated in Fig. 1, turbine blades must be cooled. Many of the newer high-temperature alloys, particularly those that can be rolled into sheet form and have sufficient ductility, make very good materials for use in cooled blades. Their high-strength capabilities make them quite useful for advanced-type engines where turbine stresses are apt to be high.

Air-Cooled Turbine Blades

Three of the cooling principles used in air-cooled turbine blades are illustrated schematically in Fig. 3. The most conventional air-cooling method used in all heat-transfer processes as well as in air-cooled turbine blades is convection cooling shown in Fig. 3(A). With this method it is desirable to increase the heat-transfer surface area on the heat-rejection side of the blade by means of fins. In film cooling, Fig. 3(B), a film of cool air is introduced through slots to form an insulating layer between the hot gases and the cooled surface. The thermal conductivity of air is very low so that it is a good insulation medium, but the effectiveness of the layer of the air is lost some distance downstream of the slot because of mixing with the hot gases. This disadvantage is eliminated by transpiration-cooling, the most effective method of air cooling known at present, Fig. 3(C), because air is continuously passed through the entire area of a porous surface.

Air-cooled blades utilizing these cooling methods, whose com-

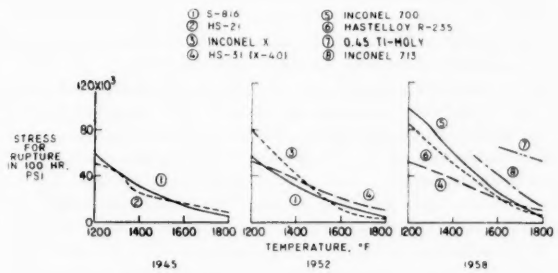

Fig. 2 High temperature turbine materials

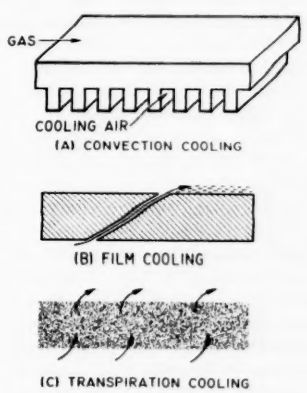

Fig. 3 Examples of several cooling methods

parative effectiveness is given in Reference [23], are illustrated in Fig. 4, with the approximate year of their development. The hollow, insert, and film-cooled blades, Figs. 4(A) to (C), were used or considered for use in engines by the Germans prior to 1945, [2], [24], [25], and [26]. The cooling effectiveness of the hollow blade is so low that excessive quantities of cooling air are required. An insert in the interior of the blade, Fig. 4(B), improves the cooling effectiveness by increasing the coolant velocity at the expense of increasing the pressure loss for a given flow rate. By the proper placing of an adequate number of slots in the blade wall, Fig. 4(C), film cooling will provide good cooling, but durability of this type of blade has been found to be poor.

The tube-filled blade, Fig. 4(D), was an early attempt of the NACA to build an air-cooled blade with additional surface area brazed in place in the coolant passage in order to improve the cooling effectiveness. Some results of tests on this type of blade and methods of manufacturing are given in [27] to [33]. Although more recent blade developments have led to superior blade configurations, much valuable information has been obtained from the tube-filled configuration, and experience was gained in the operation of air-cooled turbine blades made of nonstrategic materials. Suitable coatings for inhibiting oxidation of low-alloy steels are reported in [32].

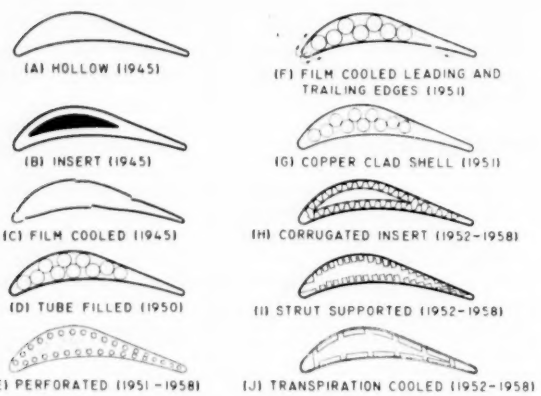
(A) HOLLOW (1945)
(B) INSERT (1945)
(C) FILM COOLED (1945)
(D) TUBE FILLED (1950)
(E) PERFORATED (1951 - 1958)
(F) FILM COOLED LEADING AND TRAILING EDGES (1951)
(G) COPPER CLAD SHELL (1951)
(H) CORRUGATED INSERT (1952-1958)
(I) STRUT SUPPORTED (1952-1958)
(J) TRANSPIRATION COOLED (1952-1958)

Fig. 4 Air-cooled turbine-blade configurations

There has been a reluctance on the part of some engine people to utilize a brazed structure in turbine blades. The perforated blade, Fig. 4(E), overcomes this objection since it does not rely on brazing. Instead, a number of holes are provided near the periphery of solid blades by drilling, casting, or other methods. Considerable research has been conducted both in England and the U. S. on this type of blade [34] to [39], which, although a more homogeneous structure, is usually heavier, resulting in higher disk stresses and generally less blade taper is feasible so that the blade stresses are higher than for some of the brazed blades.

Adequate cooling of the leading and trailing edges can be a problem with almost any configuration. In an attempt to improve cooling effectiveness in these regions of the tube-filled blade, film-cooling, Fig. 4(F), and copper-clad shells, Fig. 4(G), were investigated. The copper-clad blade is a type of structure similar to that of copper-clad kitchen utensils in which the copper spreads the heat over the entire area of the utensil. In the case of cooled turbine blades, the copper cladding is on the inner surface of the blade shell. Results of heat-transfer tests on film-cooled and copper-clad blades are reported in [40] to [42]. The effects on reduction of temperatures of turbine-blade leading and trailing edges are shown by the experimental data in Fig. 5. It will be noted that the blade-temperature gradients can be reduced by use of film-cooled and copper-clad blades, but the leading and trailing-edge cooling is done at the expense of increased temperatures in the mid-chord region of the blades. The film-cooled blade shown operated only slightly cooler at the leading edge than did the 10-tube blade. The amount of cooling obtained in this region is a function of the configuration and the relative pressures on the inside and outside of the blade. At higher flow rates, this cooling becomes more effective because a greater portion of the air is bled from the slots. The blade must be designed specifically for the application for which it is intended, and the coolant supply pressure must be higher than the stagnation pressure at the blade leading edge. More effective leading-edge cooling has been found in other film-cooled blades. Durability of this type of blade was found to be a serious problem as reported in [30]. Because of durability difficulties in turbine rotors, blades such as shown in Figs. 4(C) and (F) would probably be more successful when applied to stators, provided that cooling-air pressure is sufficiently high.

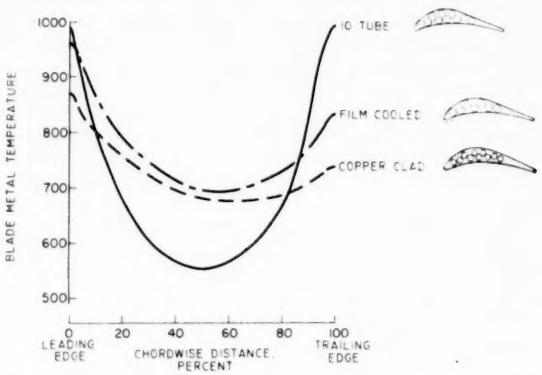

Fig. 5 Effect on metal temperature of special methods of cooling leading and trailing edges. Engine speed 10,000 rpm, effective gas temperature 1070 F, cooling-air temperature 130 F, ratio of cooling-air flow to combustion-gas flow 0.045.

Although the cooling of the copper-clad blade looks better than that for the film-cooled blade in Fig. 5, the weight increase due to the addition of copper raises the stress for rotor blades so much that the gains from cooling are practically eliminated. In addition, at the blade temperatures required in engine operation, copper oxidation is rapid and severe.

Elimination of chordwise temperature gradients in convection-cooled blades has not yet been attained and leading-edge temperatures will undoubtedly provide the limitation to permissible increases in turbine-inlet temperature. Additional research is required on leading-edge cooling for very high gas temperatures. Methods of reducing leading and trailing-edge temperatures are to use as blunt leading edges as possible from aerodynamic considerations and to use thin shells with augmented-internal heat-transfer surface, such as fins, extended well into the leading and trailing-edge regions. The corrugated insert blade, Fig. 4(H), utilizes some of these methods [43]. An island is usually provided in the middle of the coolant passage so that the corrugations can be of uniform amplitude. The core of the island is blocked off from the cooling air. Methods of designing this type of blade are given in [44].

In all the turbine blades discussed up to this point, with the possible exception of the perforated blade, the blade shell has been the primary member for carrying the stresses due to centrifugal forces. Since the shell is exposed to the gas stream, it is also the hottest member of the blade, therefore, its stress-carrying capacity

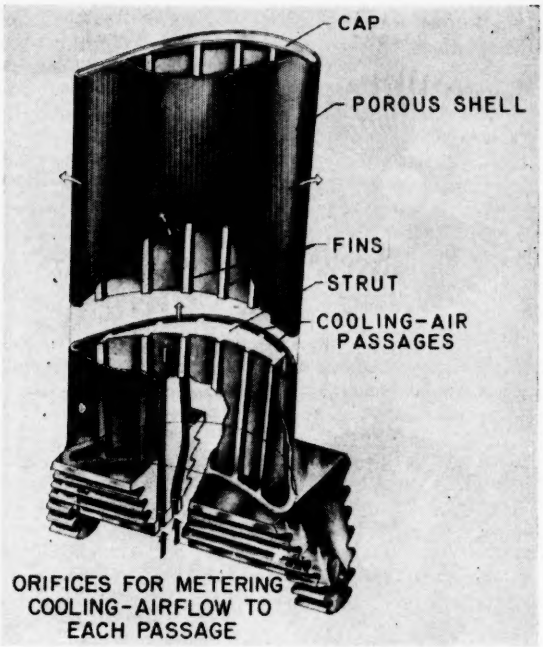

Fig. 6 Transpiration-cooled blade

is lower than that of cooler portions of the blade. For this reason it seems practical to design blades where the main stress-carrying member, or strut, is submerged inside the coolant passage, where it will be at a lower temperature than the blade shell, Fig. 4(I). This type of blade appears to be very promising for high-temperature application. Shell temperatures can be much hotter than for other types of turbine blades and lower heat-transfer rates from the hot gases to the stress-carrying member of the blade reduce coolant quantities [45] to [48].

The strut-type construction can be used with either an impermeable or a permeable shell. With a permeable (porous) shell the blade is called transpiration-cooled, Fig. 4(J). Some of the advantages and problems of transpiration cooling are discussed in [49]. The porous shell could be made from several materials, the most probable being woven-wire cloth [50] to [52], or porous-sintered materials made from powdered metal [53].

The design and manufacture of transpiration-cooled blades is difficult because the coolant-flow rate is a function of permeability of the shell and the pressures inside and outside of the blade. Because of the high-pressure gradients on the outside of the shell, a metering device is required at the blade base to meter air to each of the passages formed by the strut and the porous shell as discussed in [54]. The air can be metered by orifices as shown in Fig. 6. The shell permeability must also be controlled very carefully in order to get the proper flow distribution of the coolant that will result in uniform cooling [55].

Many analytical studies have been made on methods of calculating cooling-air requirements for the various types of blades shown in Fig. 4. Calculated coolant flows for five of the blades for a range of turbine-inlet gas temperatures are shown in Fig. 7. These flows are on a relative basis because the absolute magnitude of the flows is dependent upon many conditions such as the type of engine, size of blade, flight speed, and flight altitude. A review of blade-temperature-calculation methods is given in [56]. More specifically, the coolant flows for the hollow and tube-filled blade were calculated by methods presented in [57] to [60], the corrugated insert blade by [44], the strut blade by [48], and the porous blade by [49], [61], and [62]. For calculating heat-transfer rates in noncircular passages [63] and [64] can be used. The load-carrying member of all blades was assumed to be cooled to 1240 F.

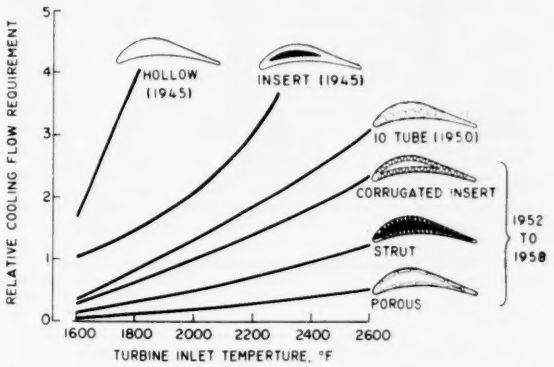

Fig. 7 Comparison of relative coolant-flow requirements for several types of air-cooled turbine blades

As shown in Fig. 7, the use of plain hollow blades is very impractical for cooled turbines because the coolant-flow requirements are exorbitant. On the other extreme is the transpiration-cooled turbine blade which ideally requires only very small amounts of coolant even for large increases in turbine-inlet temperature. In practice, however, the excellent cooling of the transpiration-cooled blade has not been demonstrated due to difficulties in obtaining uniformity of flow over the blade surface. In order to keep local areas from overheating, coolant flows have had to be high and, as a result, other areas are much overcooled. Further development in methods of precisely controlling shell permeability is required to overcome this difficulty in transpiration-cooled blades.

Besides low coolant flow, other factors that must be considered in turbine blades are fabrication problems, blade size, cooling-air pressure required, clogging (such as may be encountered with transpiration cooling), blade weight, and durability. As a result, the final choice of the blade design is up to the designer's discretion and will be dependent upon the over-all design and proposed application of the engine.

As a further help in choosing blades, studies have been made in methods of evaluating cooled-blade designs in [44], [65], and [66]. Instrumentation methods for experimental-evaluation turbine blades are discussed in [67]. The effect of blade chord on weight and cooling characteristics of air-cooled blades is presented in [68]. In general, chord has much less effect on weight for air-cooled blades than for uncooled blades and large-chord blades are easier to cool, so that in cooled turbines it is generally desirable to use as large chord blades as is feasible from structural and aerodynamic considerations.

Air-Cooled Turbine Disks

The use of air-cooled turbine blades will require a different type of turbine-rotor construction than that in current use. There is, however, considerable freedom in the type of design possible. Two main types of turbine rotors are the split disk, Fig. 8, and the shrouded disk, Fig. 9. Discussions of these disk configurations are given in [69]. With either type of construction, the cooling air can be supplied from the upstream direction, the downstream direction, or through a hollow turbine shaft. With any of these possible types of construction, internal vanes are required in the turbine rotor to direct and help pump the cooling air out to the blades. Experimental tests have been conducted on several

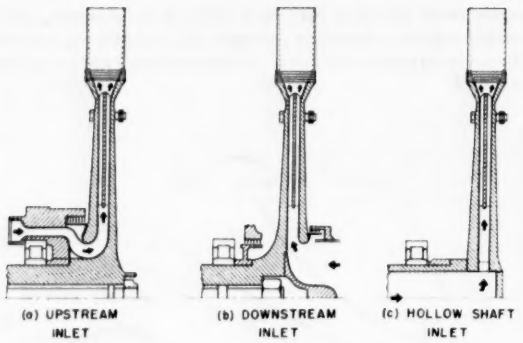

Fig. 8 Split-disk type air-cooled-turbine configurations

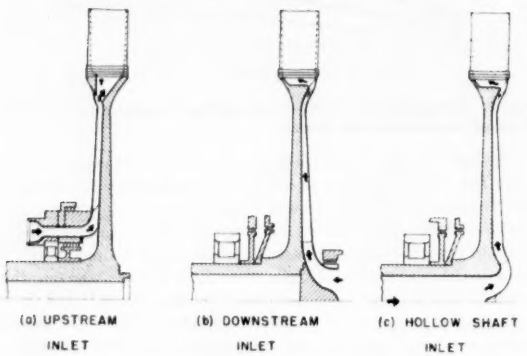

Fig. 9 Shrouded-type air-cooled-turbine-disk configurations

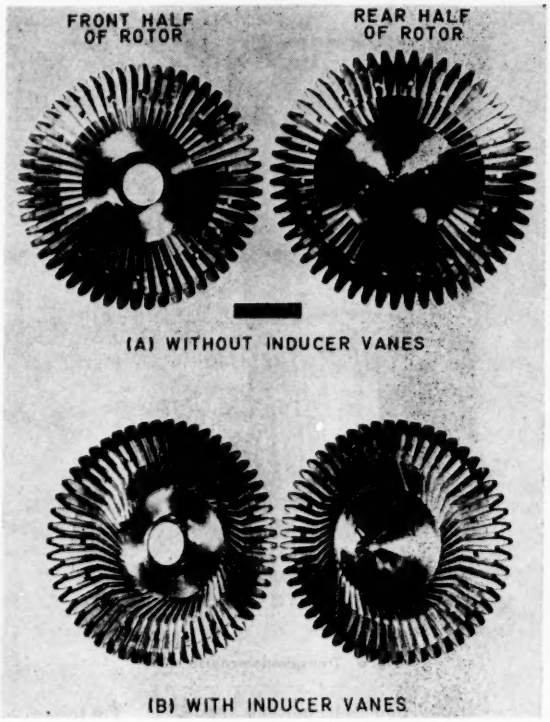

Fig. 10 Impeller vanes for air-cooled turbine disks

turbines with the type of disk configuration shown in Fig. 8(b), and some of the results presented in [70] to [72] indicate that disk cooling will be adequate with the amount of air required for blade cooling.

Fig. 10 shows two types of impeller-vane configurations that have been investigated in air-cooled turbine disks [73]. The experimental results showed that there was little or no advantage in providing the curved inducer section in the vanes as shown in Fig. 10(B). For a given coolant-flow rate, the vane configuration did not significantly affect the supply pressure required. Further, it was found that the efficiency of such an impeller system in the turbine disks has a negligible effect on engine performance.

Liquid Cooling

Internally water-cooled turbines have been operated in this country and in Europe [74] to [76], a logical method because of the excellent heat-transfer characteristics of water. Water as a turbine coolant, however, has one very serious disadvantage—the boiling point is so low that, unless the entire coolant system is under very high pressurization, the turbine is overcooled and the heat-rejection rates are excessive. A further disadvantage occurs in aircraft at high-flight speeds because the ram-air temperature exceeds the boiling temperature of water at normal pressures, and heat rejection in a ram-air heat exchanger becomes difficult. The ram-air temperature reaches 212 F at a flight Mach number of about 1.2 at standard sea-level conditions and at a Mach number of about 1.9 in the stratosphere.

Pressurization of the entire coolant system would offer relief, but in order to pressurize the system to a point where water could be at a temperature that would not result in overcooling of the turbine, the system would have to be pressurized to approximately 3000 psi. A method of eliminating this difficulty would be to use special coolants such as Dowtherm or liquid metals with higher boiling points than water at normal pressure levels.

The design of liquid-coolant systems requires a knowledge of heat-transfer coefficients for forced and free convection and the temperature gradients to be expected in the blade metal. Considerable experimental and analytical data [77] to [80], have been obtained that are applicable to liquid-cooled turbines.

The British have given thought to the use of a thermosiphon liquid-coolant system for turbine blades [81] and [82], where the small amount of coolant placed in the turbine is not enough to fill the blade-coolant passages. The coolant evaporates in the blades and the vapor is passed over a heat exchanger and condensed. The condensate then flows back to the turbine blades to be evaporated again. This system holds promise for some applications. Its primary advantage over other types of liquid-coolant systems is that the internal pressures in the blades can be considerably lower.

Water-spray cooling offers possibilities for certain limited applications where a boost in power is required for short periods of time [83] and [84]. Spray cooling for prolonged periods of time, however, results in excessive liquid consumption because the water used as a coolant is lost. Using this method of cooling the rotor blades alone permits the turbine-inlet temperature to be increased to the point where the stator blades provide a limitation.

For relatively small rotor blades, the temperatures can be reduced adequately by spraying water onto the blade from a few (up to 8) stationary circumferential locations. However, because of low water velocities relative to the gas and turbine-wheel velocities, the coolant supplied from stationary locations impinges on the blades near the leading edges only and cooling may not be provided at the trailing edges of large rotor blades. This difficulty can be overcome by supplying water spray from rotating

orifices at the blade base. A "rotating-gutter" scheme was suggested in [83]. Thermal shock to the blades when the water is introduced has been found to be a problem. Some materials are much better than others, but nearly all blades of turbojet-engine size eventually fail after some given number of shock cycles.

To increase turbine-inlet temperatures above the limit imposed by uncooled stator blades using water spray as a coolant would involve additional complications and larger total coolant flows would be required. The total flow would be about 2 to 4 times the weight-flow rate of fuel required for a single-stage engine. For multistage engines the flow weights would probably be higher yet.

Consideration has been given to fuel injection ahead of the turbine for afterburner use [85]. Although very few experimental temperature data are available, it is not believed that the quantities of fuel required for suitable afterburning temperatures would be adequate for use in cooling turbine blades. Water would still have to be injected to augment the cooling from the fuel spray.

Water-spray cooling would probably be most beneficial during take-off of aircraft having marginal take-off power. For such a case, the turbine-inlet temperature could be raised a few hundred degrees. The water used for cooling would be expended during the take-off run, and the weight of the coolant would not have to be carried during the remainder of the flight. This method of cooling would also be useful for short bursts of emergency power during flight.

Concluding Remarks

The extent to which turbine cooling will be used in gas-turbine engines is still unknown. It is clear that for certain applications it would be desirable in order to permit increased turbine-inlet temperature, while for some other applications there is no need to operate the engines at higher temperatures. For these cases when it would be desirable to cool the turbines, there are many cooling methods possible and, as evidenced by the number of References, much information is available from research that has been conducted. It is up to the engine designer to select the cooling method that is most suitable for his engine type and that uses fabrication procedures consistent with his ideas on sound construction principles.

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FUELS[1]

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A summary of the progress made since 1952 with respect to gas-turbine fuels is presented together with a review of the operating experiences reported on gaseous, liquid, and solid fuels. The technical problems involved are indicated as well as the methods employed for their solution. An appraisal of those areas requiring further developmental study is also included.

Introduction

AT THE TIME of the 1952 Gas-Turbine Progress Report, difficulties imposed by the combustion of such fuels as residual oils and coal were restricting the widespread application of the stationary gas turbine. Developmental work was needed if the gas turbine was to find commercial service irrespective of the type of fuel.

The developmental and commercial experience that has been reported since 1952 on the subject of gas-turbine fuels will be summarized and those areas requiring further study will be indicated. For convenience, the report is divided into three sections: Gaseous, liquid, and solid fuels.

Gaseous Fuels

The accomplishments of gas turbines using gaseous fuels have been outstanding during the past six years. Enviable service has been registered by a great number of units under conditions of long-term operation. There has also been a considerable increase in the number of applications involved, notably as integral parts of various industrial processes. Experience in these areas substantiates the fact that the gaseous-fuel-fired gas turbine is indeed a successful prime mover.

Gaseous fuels that have found application in the commercial gas turbine include natural, sewage, refinery, and blast-furnace gases. As these fuels generally contain only minute amounts of ash, corrosion and deposition are minimal if present at all. However, the bulk and attendant high transportation costs of many gaseous fuels limit their use to those locations in which the fuels are produced.

A high heat content of approximately 1000 Btu per cu ft together with a virtual absence of ash make natural gas ideal fuel for gas turbines. Such turbines are employed in power production and pipeline-pumping applications notably in Southern Europe, the mid-East, Venezuela, and the United States. Although space makes it impossible to describe each installation, natural-gas-fired units have generally been free from troubles attributable to the fuel and have also accumulated very satisfactory service experience. A large number of units have exceeded 20,000 hr of operation, while some have accumulated approximately 40,000 hr. About 250 gas-turbine units which use natural gas as fuel have been installed, or are scheduled to be installed, throughout the world—decidedly more units than are fired with any other fuel.

The ability to burn gases normally produced by certain chemical industries and processes has resulted in the recent adoption of the gas turbine as an integral part of these operations. In this manner the turbine can supply the heated pressurized gases, steam, or electrical power necessary for the processes involved. Approximately 20 units are operating, or are scheduled for operation, throughout the world on refinery gas with a thermal content about 500 Btu per cu ft, or sewage gases with about 700 Btu per cu ft. The wide range of thermal contents likely to be experienced with the use of refinery gases makes turbine start-up extremely difficult with a standard ignition-and-control system. Various gas-turbine manufacturers, including Clark Bros. Co., have met this problem by starting the turbines on fuels of constant heat content and then switching to the type of refinery gas available [1].[2] This appears to be a most successful application of the gas turbine, but the true assessment must await more operation and greater experience.

In Europe one of the most rapidly expanding applications for the gas turbine is associated with the steelmaking industry where unpressurized exhaust gases of the blast furnace, which contain about 90 Btu per cu ft, are compressed and burned in the gas turbine. The unit in turn supplies the hot pressurized air and/or the electrical power required for the operation of the steel mill. Approximately 20 gas turbines which will use blast-furnace gas as fuel are installed or are scheduled for the near future. These are primarily European applications although one will be placed in service in the United States [2].

Of prime importance is the degree to which the blast-furnace gas is cleaned as a significant dust content could cause fouling or erosion of the gas compressor and turbine. Erosion attributable to the fuel apparently has not been a major problem to date, fouling being the more generally observed occurrence. Stys [3] and Pfenninger [4, 5] have reported that both the blast-furnace-gas compressor and the turbine of a Brown Boveri 5400-kw unit were in quite clean condition after some 34,000-hr operation on a fuel normally filtered to a dust content between 4 and 5 mg/Nm^3 (3 to 4 ppm). The gas compressor of another unit, however, had to be cleaned at intervals of 9 weeks when a fuel containing a dust content of approximately 50 ppm was used, although in this case the turbine blades remained quite clean.

Sulzer Bros., Ltd. regard a 7-mg/Nm^3 (5.4-ppm) dust content as being the upper limit for open-cycle installations such as the 7500-kw units at Sambre, Belgium. The decrease in power resulting from fouling was not particularly high as is shown in Fig. 1, and no significant erosion could be detected after 4400 hr. Turbine-blade deposits, which varied between 0.04 and

[1] When this paper was written, the author included information and expressed opinions believed to be correct and reliable. Because of the constant advance of technical knowledge, the widely differing conditions of possible specific application, and the possibility of misapplication, any application of the contents of this paper must be at the sole discretion and responsibility of the user.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, July 18, 1958. Paper No. 58—A-46D.

[2] Numbers in brackets designate References at end of paper.

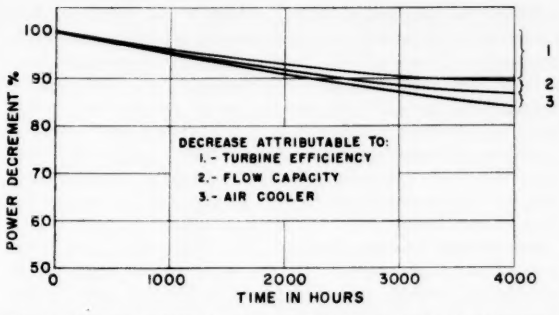

Fig. 1 Relative decrease in power attributable to dust in blast-furnace-gas fuel. Gas turbine: Heinaut 1; fuel dust content: 7 mg per Nm³ (5.4 ppm). Data courtesy Sulzer Brothers, Ltd., STH 1644.

used by Ruston and Hornsby Ltd. as gas-turbine fuel in a few British applications. The first was commissioned in 1956 and used a fuel of controlled sodium content. Fortunately, the vanadium content of creosote pitch is generally quite low [7].

One of the most significant advances has been the use of the gas turbine in crude-oil-pipeline-pumping installations, notably in the mid-East. Here the low-weight, low-water-consumption, maintenance, and manpower requirements, together with the satisfactory operation to date of the gas turbines indicate a record as successful as that realized in the natural-gas-repressurization installations. Approximately 20 of these units, including Ruston Mark TA 750/1000-kw and General Electric 5000-hp turbines have used, or will use, the crude oil being pumped as prime fuel. In both cases a magnesium-sulfate additive is used in instances where the sodium and vanadium present in the oil might cause gas-path corrosion. G-E units had

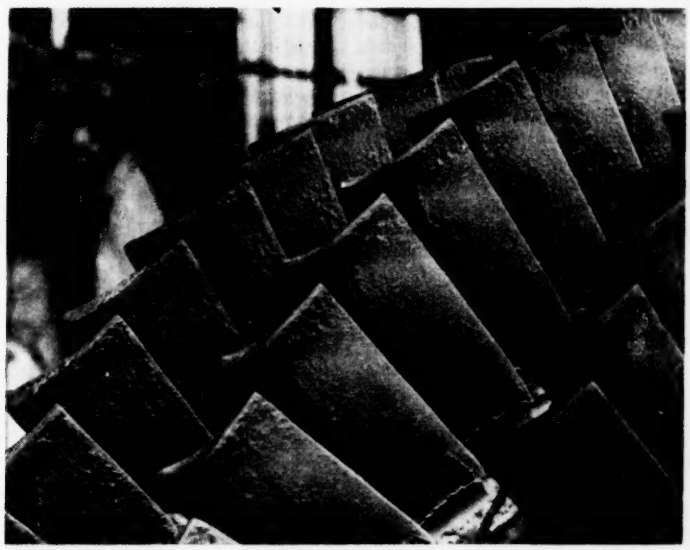

Fig. 2 First and second-stage rotor blades of 7500 kw Heinaut 1 turbine after 4400 hr operation on blast-furnace gas of 7 mg per Nm³ (5 ppm dust content). Courtesy of Sulzer Brothers, Ltd.

0.06 in. in thickness, Fig. 2, could be readily removed with a metallic brush [6].

Liquid Fuels

Two major advances have occurred since 1952 with respect to gas turbines using liquid fuels. First has been the introduction of crude-oil-fired turbines in oil-pipeline-pumping installations. Second has been the demonstrated ability of the gas turbine to burn residual fuels successfully under conditions of commercial operation.

Liquid fuels that have been used in gas turbines during the past six years include distillate, crude, and residual oils, and to a lesser extent creosote pitch. Application of fuel produced from oil shale awaits economically feasible production methods, and has not been reported in the literature.

Service experience with distillate oils, used both as prime and alternate fuels, has been almost as favorable as that with natural gas. However, competition from other prime movers has virtually dictated the use of cheaper grade fuels in most marine and industrial-gas-turbine installations. Approximately 35 distillate-oil-burning gas turbines are installed, or about to be installed, throughout the world. Creosote pitch has also been accumulated 10,000 operational hours by January, 1958. Significantly, most of these units were unattended and were remotely controlled from base stations 50 to 100 miles away [8, 9].

However, the use of residual oils is currently the key to the widespread application of the gas turbine. The present ability of both the steam plant and the diesel engine to burn residual fuels at efficiencies equivalent to, or higher than, the gas turbine compels the use of this inexpensive oil for most marine, locomotive, and industrial gas-turbine applications. Of the problems associated with the use of residual oils in gas turbines—fuel handling, combustion, corrosion, and deposition—the first two have been solved, and much effort has been devoted to the last two. Success with corrosion and deposition problems is attested to by the fact that approximately 90 gas-turbine units are installed or scheduled that use residual oils as fuel.

Residual-fuel-ash corrosion and deposition problems are by no means completely solved, and merit special attention. In some instances, corrosion is more significant, in others, deposition, and in still others, both. To generalize over the current range of practical turbine-inlet temperatures, corrosion may be appreciable at temperatures of 1150 to 1200 F and becomes more severe with further temperature increase. Residual-fuel-ash deposition, on the other hand, can be experienced throughout

the entire range of inlet temperatures, and normally increases with both temperature and pressure.

Wide differences in the physical and chemical characteristics of residual fuels, together with the variations in testing techniques make it difficult to correlate results. However, it is generally concluded that both corrosion and deposition are attributable to a molten or partially molten ash, and that the prime offenders are combinations of sodium, vanadium, and sulfur that occur in most residual fuels [10, 11, 12].

Vanadium can be present in many residual oils in concentrations up to 500 ppm and may revert to vanadium pentoxide (V_2O_5) with a melting point of 1274 F during the gas-turbine combustion process. Its corrosive action is based on the destruction of the protective oxide layer that normally exists on the surface of heat-resistant alloys, thereby exposing the base metal to attack [13, 14].

Sodium, often present in amounts up to 200 ppm can combine with sulfur to form sodium sulfate (Na_2SO_4) with a melting point of 1623 F. When suitably catalyzed by such agents as chlorine or carbon, sodium sulfate preferentially attacks the metal grain boundaries. This insidious type of attack may escape visual detection until its effects are catastrophic; the only sure means of early detection is by metallographic examination of the parts in question.

However, residual-fuel ashes containing only vanadium pentoxide or sodium sulfate rarely occur. In practice it is the interaction of these two chemicals to form complex vanadates that actually causes more concern [15]. A number of investigators have found that the most corrosive vanadium-pentoxide sodium-sulfate mixture contains between 10 and 20 per cent sodium sulfate [14, 16, 17]. This may well correspond to the formation of sodium vanyl vanadate ($Na_2O{\cdot}V_2O_4{\cdot}5V_2O_5$) whose melting point of 1157 F [18] is well within the range of gas-turbine operating temperatures.

Nevertheless there are several potential methods of minimizing the effects of residual-oil-ash corrosion and deposition to a level consistent with long-term commercial-gas-turbine operation. In view of their importance to the future of the gas turbine, the progress made in each during the past six years will be summarized.

Turbine Design. Air can be substituted for combustion products as the working medium in a system such as the closed-cycle gas turbine shown in Fig. 6. In this manner corrosion and deposition problems are transferred from the turbine to the air heater. The sacrifice so made is the need to develop a suitable air heater and the additional equipment cost thereby incurred. The 12,500-kw installations at St. Denis, France, by Escher Wyss Ltd., and at Dundee, Scotland, by their licensee, John Brown and Co. Ltd., are notable examples of the residual-fuel-fired closed-cycle gas turbine.

Most European manufacturers have minimized the effects of residual-fuel combustion products in the open-cycle gas turbine by limiting turbine-inlet temperatures to 1200 F. This results in a sizable penalty in potential efficiency and capacity. Under these conditions ash deposition is generally more pertinent than corrosion and is minimized by the use of additives or by periodic cleaning.

Of notable mention in this category is Brown Boveri's installation at Breznau, Switzerland, consisting of 13,000-kw and 27,000-kw units both with 1200 F turbine-inlet temperatures. After 11,000 hr, the turbine-rotor buckets of the 27,000-kw unit were in perfect condition, but corrosion primarily attributed to nonuniform temperature distribution necessitated partial reblading of the first row of stator vanes. Fouling decreased the power output of the 13,000-kw unit some 10 per cent over a period of 135 hr, but the original capacity could be restored by washing at 2-week intervals [4].

Efforts to use cooled blading, which would permit higher operating temperatures while maintaining relatively low blade temperatures, have still not been realized in commercial application. The Pametrada Research Station's work employing an experimental turbine with ceramic stator nozzles and liquid-metal-cooled rotor blades will be watched with keen interest. Inlet temperatures are of the order of 2200 F, and results have been sufficiently encouraging for Brown to state "one may look forward to the development of a full-scale unit in the not-too-distant future" [19].

Material and Protective Coatings. No single alloy, protective coating, or ceramic material has been found that, by itself, will be completely immune to attack by all residual-oil ashes under conditions of gas-turbine operation. The most resistant alloys available have therefore been combined with other methods of minimizing the problems associated with residual-fuel combustion products.

The wide range of testing conditions, materials, and fuels reported in the literature makes it difficult to correlate the resulting data. However, the superiority of nickel-based alloys over iron-based alloys to vanadium pentoxide has been confirmed on the basis of tests ranging up to 1600 F [20, 21, 22]. The situation with respect to ashes containing both vanadium pentoxide and sodium sulfate is less clear. This results from the fact that significant intergranular attack promoted by sodium sulfate either escapes detection or has had insufficient time to occur in many tests of relatively short-time duration. Betteridge [22] conducted a series of stress-rupture tests at 1382 F and 1598 F in which a number of alloys were coated with various vanadium-pentoxide sodium-sulfate mixtures. He concluded that nickel-based alloys, although by no means immune to attack, were in general more resistant than were either cobalt or iron-based alloys. Nickel-based alloys were, however, more susceptible to intergranular penetration as was demonstrated by the extreme scatter in stress-rupture values. This effect could become much more pronounced in tests of longer duration. That it may become catastrophic at 1600 F in nickel-based alloys when exposed to ashes of high sodium content has been vividly demonstrated by Young, et al. [11].

Alloying elements clearly have an effect on the resistance of metals to corrosion. Chromium has been found beneficial in concentrations exceeding 16 per cent [20, 23], but its gradual surface depletion by vanadium attack may render the effect only temporary. Silicon [22, 23] has also been found to impart some protection, its optimum concentration being 5 per cent. On the other hand, molybdenum and vanadium can be detrimental under certain conditions [17, 23]. Harris [20] has suggested that the vanadium and molybdenum content should be limited to 0.5 and 3.0 per cent, respectively, in alloys containing more than 30 per cent iron. In alloys containing less than 30 per cent iron the vanadium should be less than 2 per cent, while molybdenum additions will not be significantly harmful.

Protective coatings have likewise offered no complete solution during the past six years although some have shown promise under the conditions of testing. Attempts to use chromium, nickel, phosphorus, and silicon coatings, among others, have either suffered from severe attack of the coating or inability to maintain an adherent bonding between base metal and the coating [14, 17, 21, 23]. Sintered titanium-carbide ceramic blades exposed for 50 hr at 1088 F to 1460 F to a residual fuel containing 258-ppm vanadium and 9-ppm sodium displayed a small amount of localized attack; in addition, deep cracks formed on the leading edge of the blades [21].

In summary, no solution to the corrosive attack of residual fuels has yet been found by the choice of alloys alone.

Oil Modifications. Removal of the harmful elements in residual fuels, namely vanadium, sulfur, and sodium would serve as an

effective method of eliminating both corrosion and ash deposition in commercial-gas-turbine operation.

Both vanadium and sulfur are generally present in the form of oil-soluble organic complexes and therefore cannot be separated by purely mechanical means. Their removal by chemical methods has not been justified to the present time for two principal reasons. First is the complete degree of vanadium removal required, as concentrations in the order of a few parts per million are sufficient to initiate corrosion [10, 16]. Second, the methods available, even though technically feasible, would destroy the advantageous price differential that now exists between residual and distillate oils. In fact, distillation itself probably remains as the cheapest method of removing the elements. The incentives to the oil companies to develop such oil purification could be supplied in the future by the steam-turbine industry, whose gradually increasing operating temperature is presently bordering on the ranges in which corrosive attack and excessive deposition may take place in the boilers.

Sodium can be separated by mechanical means, as it is generally in the form of sodium-chloride crystals or salt-water emulsions that have originated from briney wells or tanker shipment. Such methods as electrostatic precipitation, filtration, or water washing and centrifuging can be employed. It is necessary to desalt the oil at the point of use in most applications, as desalting at the refinery would only expose the oil to recontamination during tanker shipment over salt water. The removal of sodium from residual oils was pioneered by Buckland, Gardiner, and Sanders [10]. Their work had resulted in the desalting of some 19 million gal of residual oils at General Electric installations by June, 1958. The oils ranged between 8.4 and 18.9 in API gravity. In addition to a more than tenfold reduction in sodium content, the washing and centrifuging process employed has served to reduce the dirt in the oil together with the concentration of such compounds as calcium, silicon, and iron.

Ash Separation. Removal of the ash from the combustion-gas stream is handicapped by the fineness and low concentration of the ash, and the fact that the relatively low fusion temperatures of the particles would probably result in severe slag formation in any separation equipment used. Further, the smallest particles are the hardest to remove from the gas stream and are unfortunately the most prone to cause deposition. In view of the technical difficulties no substantial data concerning the application of either electrostatic precipitators or cyclone separators to residual-fuel-ash removal have been reported in the literature during the past six years. High-temperature filter beds of various types have been employed on an experimental basis [24, 25], the prime drawback being the unacceptable pressure drop initially incurred and its subsequent increase with time. In addition the incorporation of filter beds often affords no substantial reduction in the rate of deposition.

Removal of ash products from the combustion stream therefore has not, and still does not, appear a very attractive solution for commercial-gas-turbine operation on residual oils.

Rendering Ash Products Ineffective. As no method has yet been devised that will prevent the passage of at least some ash products through the open-cycle gas turbine, the recourse left to designers is to make these ash products innocuous. Two alternatives have been presented, namely those of incomplete combustion and additives.

Incomplete Combustion. The influence that incomplete combustion has upon ash deposition was first reported by Bowden, et al. [18], and a number of other investigators have subsequently evaluated it as a means of burning residual oils in gas turbines.

"Stack solids," the unburned carbonaceous skeleton of the fuel droplets, can often be formed at the sacrifice of only 1 or 2 per cent in combustion efficiency. This carbonaceous skeleton entraps a significant portion of the ash products and can minimize deposition both by reducing vanadium pentoxide to oxides of higher melting point and by exerting a scouring effect on the turbine blades themselves. Incomplete combustion has been effective in minimizing, but not completely eliminating, deposition at temperatures lower than 1200 F. At temperatures higher than 1200 F, however, it has proved less effective in reducing ash deposit and apparently may even intensify the corrosive attack of residual fuels [11, 18, 25].

The disadvantage of incomplete combustion lies primarily in its production and control; at least one instance having been reported where it could not be produced to a significant extent even with the most unfavorable combustion characteristics [26]. Incomplete combustion is generally produced by employing coarse fuel atomization, short combustion-residence time, or low combustion temperatures. The control of any of these has thus far been judged too difficult for commercial-gas-turbine application, especially in cases where fuels of widely different physical properties might be encountered.

Additives. Most gas-turbine designers have employed additives as a means of overcoming the problems associated with the combustion of residual fuels, irrespective of the turbine-inlet temperature involved. It is in this field that a considerable amount of work has been conducted during the past six years. Additives, however, cannot be economically justified if they absorb a substantial portion of the price differential presently existing between distillate and residual oils.

Many additives exist in oil-soluble, water-soluble, and solid forms, generally decreasing significantly in both cost and ease of handling in that order. A given additive appears to be equally effective in any of these forms if it is intimately mixed with the oil. In the case of solid additives, it must be of sufficiently small particle size so that each individual fuel droplet is correctly treated. Walker's [27] mathematical treatment of the latter aspect is a classic in the field.

A great number of additives have been examined with varying degrees of success. Those that have been used extensively in commercial-gas-turbine applications are of immediate interest and include magnesium, aluminum, and silicon, either singly or in combination.

Aluminum or silicon-based additives are only partially effective in preventing the corrosive attack of residual fuels and therefore cannot be used alone at temperatures above 1200 to 1250 F. They may, however, reduce turbine fouling substantially over that for the untreated fuel although the mechanism by which this occurs is not clear. Aluminum additives have not resulted in a substantial decrease in deposition when used singly [18, 25], but have been used advantageously in combination with other additives. For example, aluminum markedly reduced the fouling rate of a magnesium-based additive at a turbine-inlet temperature of 1500 F [28]. Silicon has decreased deposition substantially at temperatures below approximately 1250 F, but is not as effective at higher temperatures [18, 24, 25, 26]. Inexpensive silicon additives are all of the insoluble-powder type and introduce considerable handling difficulties. One such powder, aluminum silicate or kaolin, has reduced turbine fouling greatly in some gas-turbine applications [24, 26, 29].

Aluminum silicate has been used in the 20,000-kw semi-closed-cycle Weinfelden installation of the Sulzer Bros., Ltd. with turbine-inlet temperatures up to 1256 F. It has also been applied to the 1200-hp British Thomson Houston open-cycle unit aboard the Shell tanker *Auris*, operating temperatures in this case being up to 1200 F.

The *Auris* gas turbine, Fig. 3, operated 6649 hr out of a total 20,510 hr on residual oils mostly containing between 63 and 300-ppm vanadium and 44 to 87-ppm sodium. Aluminum silicate added to the fuel in concentrations above 0.05 per cent substantially reduced ash deposition in some instances, but was hard

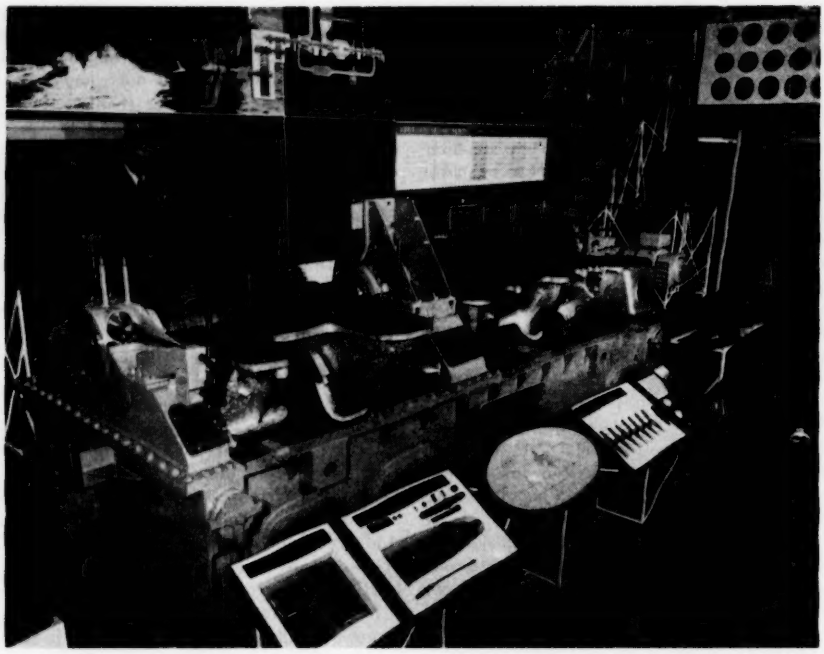

Fig. 3 General view of high-pressure turbine and compressor of 1200-hp British Thomson Houston gas turbine after 20,510 hr operation on the Shell tanker *Auris*. The low-pressure turbine is at the extreme right. Courtesy Shell Petroleum Company, Ltd.

to handle aboard ship because of its extreme fineness (approximately 3-micron particle diam). Corrosive attack was quite severe on the original iron-based Rex 337A turbine blades after 11,372 hr. However, replacement Nimonic 80 blades were virtually unaffected as a result of the final 9137 hr of operation, 4903 of which were with residual fuel. It was concluded that a useful life of at least 20,000 hr could be expected under these conditions. This experience together with the successful operation of the turbine has fully justified replacement of the original 1200-hp unit with one rated at 5500 hp that will serve as the sole propulsion system of the ship. Since the reduction in deposition afforded by aluminum silicate could largely be attributed to mechanical cleaning rather than a chemical effect, airborne cleaning compounds will be used instead to minimize turbine fouling in the new unit. Crushed walnut shells of 20/40 mesh have been used successfully for this purpose at an anticipated cost of less than ½ per cent of the fuel and are, furthermore, easier to handle than is the aluminum silicate [29].

The Weinfelden plant, which is no longer in operation [6], used kaolin with "1 to 2 per cent of metallic compounds added." A total additive dosage up to 0.45 per cent was employed in a fuel containing 100-ppm vanadium and 20-ppm sodium. Sulzer [24] reported that turbine corrosion and ash deposition were both insignificant after 2400 hr of operation on the treated fuel. In fact, turbine cleaning was never required. Air-heater corrosion was likewise overcome, and the deposits which did form in this region were readily removed by brushing at 6-week intervals. The treatment represented 3 to 5 per cent of the fuel cost, but it was anticipated that eventually this could have been reduced to between 1 and 2 per cent [30].

The effectiveness of magnesium as a corrosion inhibitor has been verified by a number of investigators [10, 11, 16, 26, 31]. Its particular effectiveness against vanadium attack has been most often attributed to the formation of magnesium orthovanadate ($3MgO \cdot V_2O_5$) with a melting point of approximately 2000 F [32]. Even when this particular compound is not found in the gas-turbine deposits, the resulting ash appears to be innocuous and probably minimizes corrosion by its blanketing effect.

Magnesium has afforded complete protection up to 1600 F against residual oils of high vanadium content (350-ppm vanadium, 7-ppm sodium) when a magnesium-to-vanadium weight ratio of 3 was employed [33]. The influence of magnesium additives upon ash deposition is not as clear, primarily because of the many testing conditions and fuels used. For example, pressure exerts a strong effect in this case. High operating pressures will reduce decomposition of the ash thereby promoting the formation of heavier sulfate deposits. The deposits resulting from the use of magnesium additives are generally not as heavy as those of the untreated fuel and are far less tenacious. In nearly all instances the deposits can be easily removed by water injection, compound cleaning, or even by machine shutdown [6, 26, 28, 33].

Magnesium-based additives have been used in commercial gas-turbine operation in units manufactured by the General Electric Company. A total of 35 units had accumulated some 550,000 hr operation on residual fuels by June, 1958, using relatively high turbine-inlet temperatures between 1350 F and 1500 F. The necessary control of both corrosion and deposition is effected in all cases by desalting the oil, if necessary, and then using a magnesium-based additive to inhibit vanadium attack. A maximum of 10-ppm sodium or a sodium-to-vanadium weight ratio below 0.3, whichever is less, has been found necessary to limit corrosion and deposition attributable to sodium. A magnesium-to-vanadium weight ratio of at least 3 has similarly eliminated vanadium attack. The expense incurred by such treatment will naturally depend upon the oil in question, but generally will be between 2 and 8 per cent of the fuel cost.

The effectiveness of the aforementioned treatment has been demonstrated by a fuel that originally contained high concen-

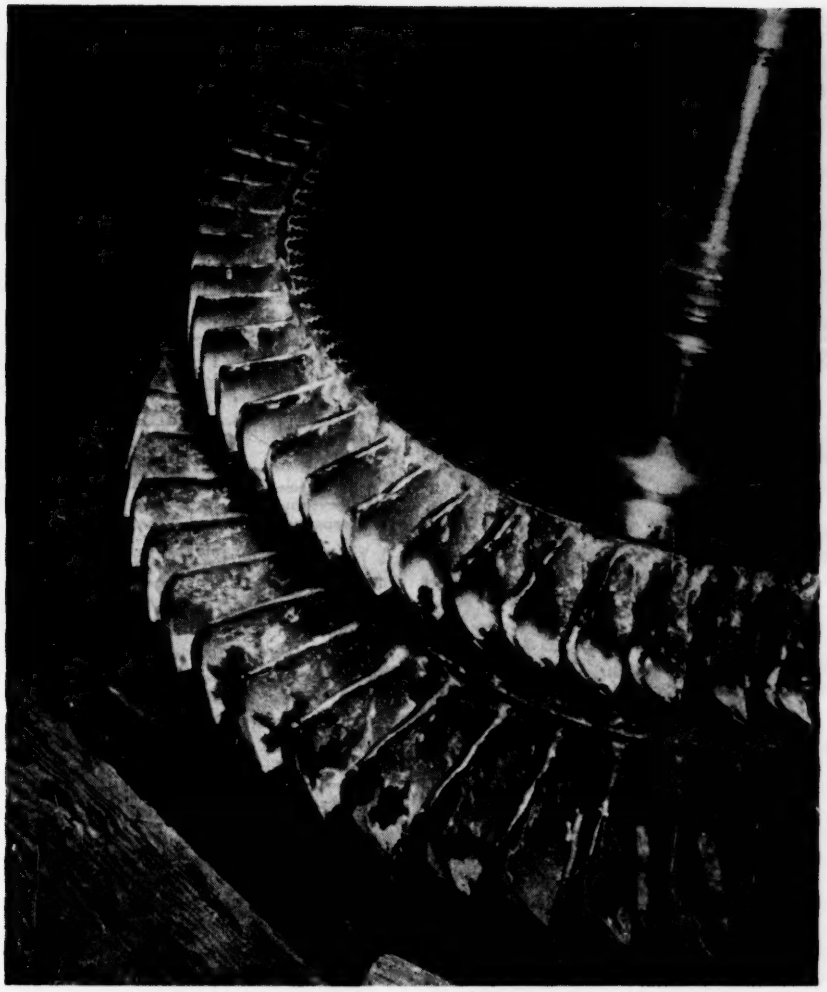

Fig. 4 First and second-stage turbine rotor blades of 5000-kw General Electric unit. Condition after 2368 hr operation on treated high-vanadium fuel preceded by 3070 hr on No. 2 distillate oil. Courtesy Bangor Hydroelectric Company, Bangor, Me.

trations of both vanadium and sodium (360 ppm and 70 ppm, respectively). After 2368 hr of operation with a turbine-inlet temperature of 1500 F, corrosion was no worse than with a treated low-vanadium residual fuel, and not much greater than that observed with the combustion of natural gas. The resultant ash reduced the output load between 0 and 13.0 per cent in 100 hr of continuous operation. It was mostly dislodged upon machine shutdown, but could also be removed by the injection of a cleaning compound [28]. The condition of the first and second-stage turbine-rotor blades after 2368 hr of operation with treated residual fuel preceded by 3070 hr on No. 2 distillate oil is shown in Fig. 4.

The 6600-hp unit shown in Fig. 5 installed on the Liberty ship *GTS John Sergeant* accumulated 5266 hr of operation by May 15, 1958, with a turbine-inlet temperature of 1450 F. Of this, 3214 hr were logged on a residual fuel that originally contained 58-ppm vanadium and 15-ppm sodium, and was treated in the manner previously indicated. After 900 hr of operation on the treated residual oil, which was the time of the last inspection, no corrosive attack could be detected throughout the gas path. Ash deposition occurring on the turbine blades and in the regenerator increased the ship's over-all fuel rate only about 3 per cent in 240 hr, and was readily removed by water washing at the completion of each transatlantic voyage [33].

It is apparent that the problems associated with use of residual oils in gas turbines have not been completely solved. However, the advances in the past six years have been sufficient to justify commercial application. The most useful solution developed to date appears to be the use of a magnesium additive in conjunction with a desalted oil. This results in almost complete elimination of corrosion even at the highest turbine-inlet temperatures presently practical. Ash deposition in this instance is tolerable and is readily removed by turbine shutdown or cleaning. If extended continuous operation is necessary, deposition can be controlled by injecting suitable cleaning compounds under load [29].

Solid Fuels

The ability to use solid fuels would substantially increase the

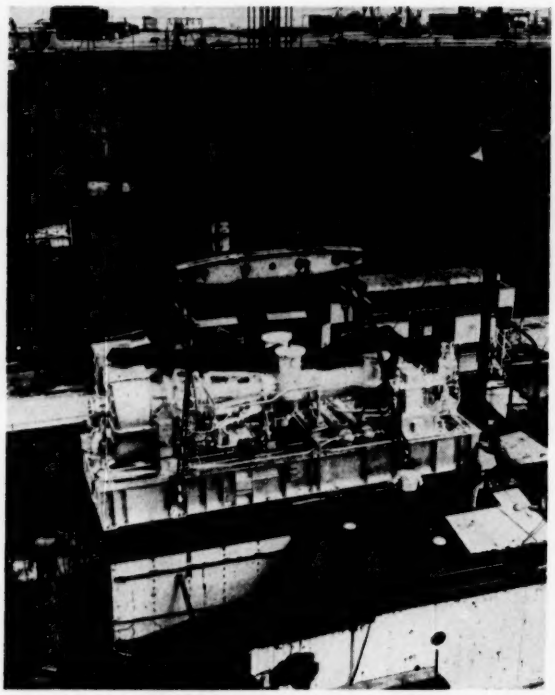

Fig. 5 A 6600-hp General Electric Company gas turbine being lowered into the Liberty ship *GTS John Sergeant*. Courtesy Newport News Shipbuilding and Drydock Company.

number of applications available to the gas turbine. It would allow the production of power in geographic regions where such solid fuels as coal and peat are domestically produced and less expensive than imported oil or gas fuels. It could also cause the advent of the combined steam-turbine gas-turbine coal-burning cycle. This cycle offers a marked gain in heat rate over that of the standard steam plant and would consequently find widespread application throughout the electric-utility industry.

Before the potential applications become a reality it is necessary that the gas turbine demonstrate its ability to burn solid fuels, notably coal, in a manner consistent with long time commercial operation. The problems involved are difficult but the incentives are great enough to justify extensive development work.

The main problems associated with the use of solid fuels for long-time commercial operation are erosion, deposition, and corrosion resulting from the ash products, any one of which may be predominant depending upon the fuel and turbine in question. There are a number of approaches available for developing a solid-fuel-fired gas turbine. Open, exhaust-heated, and closed cycles, Fig. 6, have been tested and each possesses its own distinct advantages. In addition, the possibility of gasifying coal to provide a gaseous fuel suitable for combustion has been studied.

Open Cycle. The simplest and most compact gas-turbine system is the open cycle. Fuel is burned in a pressurized combustion chamber and the resulting combustion products are expanded to atmosphere through the turbine. The technical difficulties associated with the use of solid fuels in the open-cycle gas turbine are quite formidable: Metering fuel into a pressurized combustion chamber, burning such fuel at continued high efficiencies, and removing the harmful ash products, or at least rendering them noncorrosive and in a condition to cause no deposit or erosion. During the past six years the fuel-handling and combustion problems have been solved sufficiently well

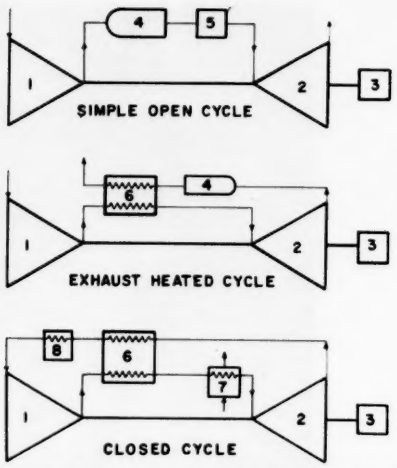

Fig. 6 Schematic diagrams of open, exhaust-heated, and closed gas-turbine cycles: 1 Compressor, 2 turbine, 3 load, 4 combustion chamber, 5 ash separator, 6 heat exchanger, 7 air heater, 8 cooler

to permit commercial application. The erosion-and-deposition phenomenon remains unsolved. This aspect is unfortunately the hardest to define as it is dependent upon many variables including temperature, fuel composition, and turbine characteristics. It therefore can only be solved completely by a combination of actual turbine experience and, most important, an investigation of the basic mechanisms involved.

Extensive tests on the coal-burning open-cycle gas turbine have been carried out by the Locomotive Development Committee in Dunkirk, N. Y., and the Aeronautical Research Laboratory in Melbourne, Australia. Different British manufacturers, mostly under the sponsorship of the Ministry of Power, have also conducted well-co-ordinated investigations on the various solid-fuel-fired gas-turbine cycles.

The Locomotive Development Committee has operated a 4250-hp open-cycle gas turbine on bituminous coal for a total of 3760 hr [34]. This consisted of three test periods, with each period displaying a considerable improvement in coal-handling techniques, combustion, and blade life. A film-cooled metal combustion chamber was evolved that maintained combustion efficiencies consistently above 95 per cent throughout the last 1100-hr test. Separation of the larger ash particles was effected in a relatively compact space, with no malfunctioning of the Mark 111D Dunlab separator tubes being detected during the same test. Turbine-blade erosion appears to be a more pertinent problem than fouling under these conditions. The work of the Locomotive Development Committee is presented in more detail in the paper by Broadley [35] in this report.

The British Ministry of Power has investigated the technical feasibility of coal-burning gas turbines [36], coal being only slightly more than half as expensive as oil fuels in the British Isles. Voysey [37] reported that slightly less than 1000 hr of operation on coal was accumulated on a 500-hp experimental gas turbine, practically all being with pulverized fuel, dry-burning-type combustion chambers. Erosion was insignificant as a result of the relatively low gas velocities involved, whereas deposition was sufficient to reduce the turbine flow area by a few per cent. It was felt that larger chord blading with the same pitch-chord ratios would minimize erosion and deposition to give acceptable running times. The importance of particles of extremely small diameters was also studied. Specifically, a decrease in deposition was effected during one test by removing a sizeable proportion of the coal particles with diameters less than 10 microns.

Analogous work sponsored by the Ministry sought to determine the applicability of peat as fuel for an open-cycle gas turbine. Peat is the only indigenous fuel in certain parts of Northern Scotland and Ireland. The gas turbine is, moreover, potentially suited for power generation in these geographical regions because of its light weight, low water consumption, and ready installation. As the moisture content of peat may exceed 90 per cent, a system of filter presses was used to remove more than half of the water. The remaining moisture removal required for fuel combustion was accomplished by using the drying action of the turbine exhaust. As a result of initial operation with peat as fuel, it was found that erosion and deposition could be experienced in a manner analogous to that associated with coal-fired turbines. In addition corrosion could also be appreciable, being attributable to such compounds as sodium and calcium sulfates contaminated with chlorides that existed in the peat ash [38, 39].

Still another method of utilizing solid fuels in an open-cycle unit consists of gasifying the fuel prior to its combustion in the gas turbine. The prime advantage is that ash removal may be effected in a relatively small volume of gas before it is diluted with air to the volumes demanded by the gas turbine. An obvious disadvantage is the additional equipment required for the gasification step with the attendant increase in cost. The general experience accumulated would indicate that solid-fuel gasification, although technically feasible, may result in a fuel cost nearly equal to that of other more suitable gas-turbine fuels. For most applications, therefore, gasification is not commercially applicable at the present time.

It is significant to record that the development projects sponsored by the British Ministry of Power have recently been discontinued. This occurred because many of the incentives have been removed rather than from technical infeasibility of the solid-fuel-fired gas turbine. Of paramount importance has been the progress of nuclear energy in Great Britain [37].

As is the case with many geographical regions, coal is Australia's only proven indigenous fuel. The Aeronautical Research Laboratory in Melbourne is presently determining the feasibility of deriving power from the immense brown-coal deposits by means of an open-cycle gas turbine [40]. Preliminary tests were conducted on an aircraft turbosupercharger in conjunction with a pulverized-fuel, film-cooled metal-liner combustion chamber. Ash separation was judged infeasible as the majority of ash particles produced were less than $^1/_{30}$ micron size, the small portion of agglomerates being approximately $^1/_3$ micron in diam. No signs of erosion could be detected, but turbine fouling was unacceptable and increased markedly with inlet temperatures between 900 and 1500 F. Ash deposition could, however, be minimized by the controlled injection of water or sand into the turbine.

It was recognized that conditions may be entirely different in a full-size gas turbine where gas velocities some three times higher than in the test rig may even reduce ash deposition because of their abrasive action. For example, Keeble [41] found that deposition on a cylindrical specimen became negligible with gas approach velocities between 700 and 800 fps, no signs of metal erosion being detected. As a means of continuing the investigation, a 1000-kw open-cycle gas turbine has been installed for operation on brown coal in the near future.

There are areas of striking similarity between the combustion of coal and residual oils, despite the wide difference in their physical properties. Removing 80 per cent of the high sodium and chloride content of one brown coal by water washing reduced the fouling rate considerably. In another test, a substantial decrease in deposition occurred after a finely ground kaolin (aluminum-silicate) "additive" was mixed with the pulverized-brown-coal fuel [40, 41].

Exhaust-Heated Cycle. The exhaust-heated cycle represents another approach to the use of solid fuels in gas turbines. Its prime advantages over the open-cycle system are that air and not fuel-combustion products are used as the working medium, and that combustion takes place at atmospheric pressure. Turbine-blade erosion and fouling are eliminated, and coal-handling problems minimized. Against these advantages is the necessity to develop an air heater that will withstand the high temperature fouling and corrosion involved and the attendant increase in equipment size and cost.

McGill University, under the sponsorship of the Canadian Department of Mines and Technical Surveys, and the C. A. Parsons and Company Ltd., have been developing this cycle. The latter company is engaged in building an exhaust-heated gas-turbine locomotive, described in another part of this report [35].

The Canadian Department of Mines-McGill University project utilized a 500-hp aircraft gas turbine in conjunction with a slagging cyclone-type furnace. Mordell [42] reported that the technical feasibility of the coal-fired exhaust-heated cycle had been demonstrated after some 1000 hr operation by November, 1956, but that some modifications would be advantageous for commercial operation. For example, the use of a cyclone slagging chamber was accompanied by relatively high heat and power losses, difficulty in control, and high part-load fuel consumption. For these reasons a pulverized-coal dry-burning-type chamber might prove more acceptable in most applications. The substitution of an air-tube heater would offer both thermodynamic and mechanical advantages over the gas-tube exchanger used. Of the greatest importance was the need for alloys that would provide economic tube life under air-heater operating conditions. The project has recently been discontinued, with the feeling that the next logical step would be the commercial development of this cycle by industry [43].

Closed Cycle. Still another approach to the use of solid fuels in gas turbines is that of the closed cycle in which, as in the exhaust-heated cycle, air is the working medium. Again, turbine-blade erosion and fouling are eliminated at the sacrifice of developing a suitable air heater. The entire system is closed and pressurized, thus permitting relatively high efficiencies and components of small size.

Escher Wyss and their licensee, John Brown Company Ltd., have both conducted extensive work on the closed-cycle gas turbine. Escher Wyss installed a 2000-kw pulverized-coal closed-cycle gas turbine in the Ravensburg plant in 1955, a turbine-inlet temperature of 1220 F being used. Some 2000 hr of operation were accumulated on the unit by December, 1956. It was found that air-heater deposits could be controlled by soot-blowing at intervals of 6 to 8 days. These deposits became partially molten and tenacious if left on the tubes for longer than one week [44]. Keller [45] reported that comparable coal-fired closed-cycle units, one of 6600 and two of 10/12,000 kw have been installed or will be installed in the near future.

John Brown has run a 500-hp experimental gas turbine for 1000 hr on both peat and coal fuel. Some corrosive attack occurred in the air heater as a result of operation with peat and was attributed to the presence of such compounds as sodium and calcium sulfates contaminated with chlorides that were present in the peat ash. The air-heater tubes were not attacked when coal was used with the exception of mild sulfur attack in one nickel-based alloy tube. On a larger scale, a 1000-kw coal-fired unit was installed in 1955 and supplies both hot water and electricity for the company's factory. In addition, two units rated at 2000 kw apiece, are installed in Scotland with peat and coal-fired air heaters, respectively [36, 39, 45].

It can be seen that the technical problems associated with solid-fuel-fired gas turbines have by no means been completely resolved during the past six years. In spite of considerable progress, the

only system that can be considered as being in commercial operation is the closed-cycle turbine. All other projects that are still active fall into the category of experimental or developmental power plants.

Summary

An objective summary of the progress that has been made during the past six years with respect to gas turbine fuels has been attempted. Those areas have been emphasized in which developmental work was, and in some cases still is, required. Space limitations have prevented the detailed description of the many areas characterized by less difficult operation.

Gas-turbine fuels fall into two main categories: Those containing virtually no ash, and those containing a significant ash content. Minimum-ash fuels have generally fulfilled expectations, but no magic panacea has appeared that will permit the unpenalized use of fuels with a substantial ash content. Turbine designers have primarily endeavored to eliminate the destructive effects of these fuels, for example, corrosion by residual-oil ash and erosion by coal ash. The less catastrophic effect, that of deposit, with its attendant loss in turbine efficiency and capacity, has then been met by the use of additives and cleaning compounds.

The suitability of most gaseous fuels for gas-turbine applications has been re-emphasized during the past six years. The technical feasibility of using virtually any liquid fuel, including residual oils, has also been demonstrated. Economics has therefore become the major determinant in this case, especially in view of the competition afforded by other prime movers. Solid-fuel-fired gas turbines remain primarily in the developmental stage, and it appears that the coal-burning gas turbine now occupies about the same status that was held by the residual-oil-burning turbine in 1952.

Acknowledgment

The author wishes to express his sincere appreciation to B. O. Buckland and R. M. Junge of the General Electric Company Gas Turbine Department, for their assistance in reviewing the manuscript of this paper.

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DISCUSSION

L. L. Newman[3]

The author has thoroughly covered the available fuels for application in gas turbines. In this connection I wish to mention a new gas turbine installation that has recently been put in operation near Tula, some 100 miles south of Moscow in the USSR. I cannot quite make up my mind whether the fuel for this turbine should be classified as gaseous or solid, because the fuel is produced by the underground gasification of coal.

Gasification is accomplished by sinking parallel rows of boreholes 25 meters (about 80 feet) apart. Air is blown into the burning seam of coal through one row of boreholes and the gas is removed through the other row. The gas is cleaned as it passes through cascade scrubbers and electrical precipitators and is cooled to a temperature of 30 C (86 F) prior to combustion in the gas turbines.

The calorific value of the gas entering the combustion chambers is 900 kilocalories per normal cubic meter (about 95 Btu per cubic foot). The initial temperature of the combustion gases entering the turbine is 650 C (about 1200 F). The turbines are rated at 12,000 kw each. They are type GT-12-3 and were developed for low-calorific value-gas produced by the underground gasification of coal by the Metalworking Plant of Leningrad. Each gas turbine assembly has two shafts, one for high pressure and the other for low pressure.

Two turbines are installed in the plant. When operating at the combined capacity of 24,000 kw, they require from 88,000 to 93,000 normal cubic meters of gas per hour (from about 3,300,000 to 3,500,000 cubic feet per hour). These gas-input requirements are expected to consume about 400,000 tons of coal annually.

The establishment is known as the Shatsk Underground Gasification Station and Power Plant. It is in the Shat brown coal basin which has an estimated reserve of 30 million tons.

A diagram of the underground gasification process and photographs of one turbine (with cover removed) and several views of the station appear on pages 10 and 11 of *Soviet Union*, No. 3(85), March, 1957. Additional technical information is given in a paper, "The Shatsk Underground Gasification Station—A New Type of Power Undertaking," by Ya Kryuchkov, in *Podzemnaya Gasifikatsya Ugley*, No. 4, 1957, pp. 3–7. A translation of this paper, prepared by the Branch of Bituminous Coal Research, Bureau of Mines, U. S. Department of the Interior, is deposited in the John Crerar Library, 86 East Randolph St., Chicago 1, Ill.

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Author's Closure

The author would like to thank Mr. Newman for his valuable contribution to the paper. As technical information on Soviet projects is not readily available, the publication of Mr. Newman's comments is most pertinent. It is hoped that more information will become available in the future on the experience gained during operation with this Soviet project.

CYCLE COMPONENTS

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Since 1952, the development of gas turbines has pointed steadily in the direction of greater simplicity, particularly in the arrangement of components in the cycle. Reheat, intercooling, and even recuperation tend to be less used than before, and only where their adoption is well justified. Experience has proved that complicated cycles, which show remarkable advantages on paper, do not work as well as expected in practice.

SINCE the publication of the 1952 Report, the development in gas turbines has pointed steadily in the direction of greater simplicity. This is particularly true of the arrangement of components in the cycle. Reheat, intercooling, and even recuperation tend to be used less than before, and only where their adoption is well justified. Experience has proved that complicated cycles, which show remarkable advantages on paper, do not work as well as expected in practice. Sometimes components are unreliable, sometimes some unexpected losses nullify the advantages. The more complex cycles seem also to take much longer before their development is complete.

This tendency toward simplicity may not be a lasting trend. When more types of gas turbines become commercially successful and when manufacturers acquire more experience and skill in developing and matching components, designers may revert to more complex cycles to obtain better results. Although the commercial development of the gas turbine has been more satisfactory than expected, the design, manufacture, and development of gas turbines is still pioneering work, and their sale meets severe psychological blocks in addition to the legitimate competition from other prime movers. Also, a successful gas turbine can only be obtained under present conditions if the main components are perfectly matched. Consequently, the manufacturing costs are very high and, once the main lines of the design have been decided, it is very difficult to introduce any relevant changes. On the other hand, commercially available components can be used in steam plants and in Diesel engines.

For the reasons given, none of the existing gas turbines is as good as the sum of the best available components would be. If such a gas turbine were built, the components were well matched, and the mechanical design was as good as in the best existing machine, considerable progress would be obtained without any new component development.

It must be remembered that most of the commercially successful existing gas turbines were designed some time ago, often before 1950. This is true both of industrial and of aviation machines. Many of the successful industrial gas turbines in the U. S. and in Britain were originally designed by manufacturers then mainly engaged in jet development, and were based on designs perfectly adapted to aircraft needs, but hardly justifiable on industrial machines. On the other hand, Swiss and other Continental manufacturers have tended to lean too heavily on industrial steam-turbine design, neglecting the advantages of lightweight construction. Usage has often frozen unsuitable designs.

Much of the trouble met in development of the more complicated cycles has been due to miscalculation of pressure losses, particularly in ducting and elbows. When sufficient experience is obtained in evaluating and reducing pressure losses, some of the more complex cycles may gradually reappear, particularly for high powers.

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INDUSTRIAL GAS TURBINES

Currently, successful commercial gas turbines seem to follow a set pattern. An axial compressor is followed by either a single-shaft or split-shaft axial turbine with the simplest bearing arrangement, and the pressure ratio is chosen so that an optional, moderately effective heat exchanger can be used. Consequently the cycle-pressure ratio is a compromise between the high values preferable in the nonrecuperative cycles and the low values required for a good utilization of the heat exchanger. Even when provision is made for an optional heat exchanger, the number of machines sold with heat exchanger is a small percentage of the total. This general pattern is followed by most of the successful medium and large industrial gas turbines, among which are General Electric, Westinghouse, Ruston and Hornsby, and the smaller and medium Brown Boveri models. The United States models generally use multiple, symmetrical, can combustion chambers, while the European types use single combustion chambers.

This trend toward simplicity will probably become even more marked in the near future. Many machines are of the split-shaft type, which gives slightly better part-load efficiency because a variation in turbine-inlet temperature changes not only the cycle temperature, but also the speed of the compressor turbine and consequently the pressure ratio, and the mass flow. Single-shaft machines are more often used with electric generators, which are essentially constant-speed machines. Apparently in the near future some successful split-shaft machines will become single-shaft machines. The reduction in part-load efficiency will be made up by lower cost, as fewer bearings are needed, and by greater simplicity and reliability.

The 20,000-kw English Electric gas turbine, Fig. 1, represents one of the few radical departures from the usual arrangement. In this machine, two similar h-p units, each complete with compressor, combustion chamber, and split-shaft turbine are used in parallel. One of them drives the generator, the other drives the large l-p compressor with intercooler, which provides the l-p air for both units. The basic h-p units are similar to the well known English Electric 2700-hp gas turbine, but more turbine stages are added to take care of the much greater expansion ratio. The h-p units have a pressure ratio of 4.5 while the l-p compressor gives a ratio of about 3.5 so that the total expansion ratio in the turbines is over 16 to 1, sufficient to give a very good efficiency without a heat exchanger. This plant is running, but the actual practical results in operation have not been published. It has the advantage of using tested components, and the h-p units are very compact for the power developed. On the other hand, there are two small high-temperature turbines, probably more expensive than a single large one, and regulation is more complicated because the temperature of the two combustion chambers must be carefully adjusted to obtain good matching between the

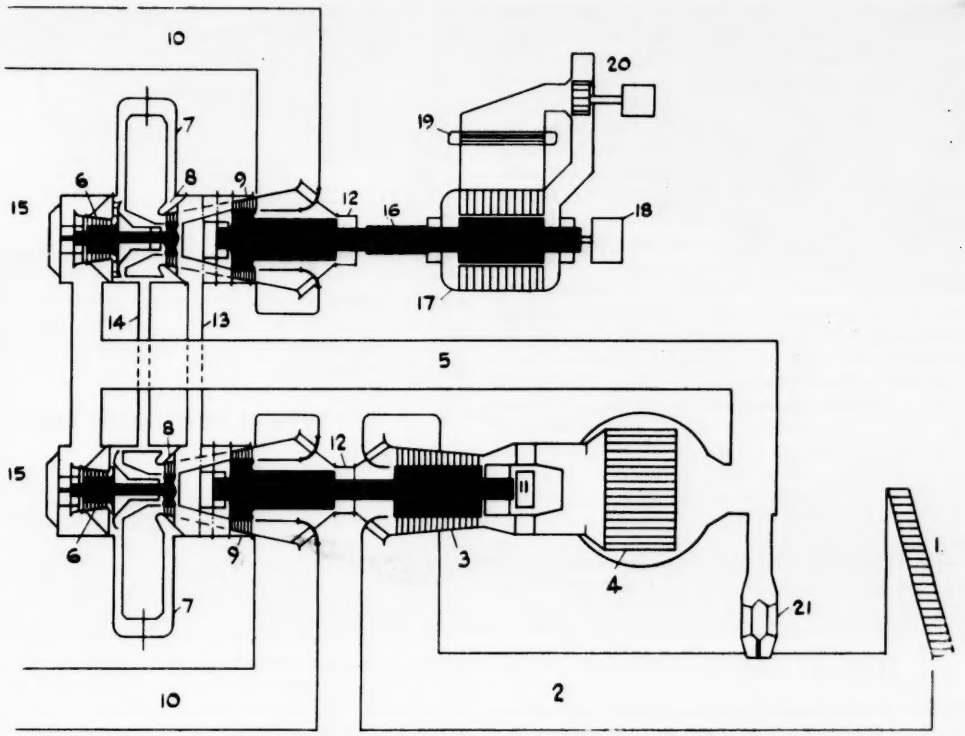

Fig. 1 20,000-kw English Electric gas turbine: 1 intake filters, 2 inlet duct, 3 l-p compressor, 4 intercooler, 5 l-p duct, 6 h-p compressors, 7 combustion chambers, 8 h-p turbines, 9 l-p turbines, 10 exhaust, 11 hydraulic barring gear, 12 thrust block, 13 l-p balancing duct, 14 l-p balancing duct, 15 h-p gear boxes, 16 drive shaft, 17 alternator, 18 l-p gear box, 19 alternator 20 alternator fan 21 l-p blowoff valve (Courtesy English Electric Company Ltd.)

two sets in parallel. Commercial acceptance or rejection will be the test of this type of machine.

The tendency toward simplicity has also prevailed in closed-cycle gas-turbine plants. When the 1952 Report was being prepared, the Escher Wyss 12,500-kw plants for St. Denis and Dundee were starting their tests. These plants were extremely complicated, with several stages of compression, intercooling, reheat, and even a separate open-cycle gas turbine to supercharge the combustion air. The results are not satisfactory. The efficiency is lower than expected and the combination and matching of so many components has caused much trouble. After this experience with an extremely complex cycle, Escher Wyss went over to a very simple machine, illustrated in Fig. 2 [1].[1] The turbomachinery consists of a 3-stage centrifugal compressor, with intercooling, and a 5-stage axial turbine combined in a single casing with the precooler. Several 2000-kw units have been built; a complete plant is running at Ravensburg, Germany, with a coal-fired air heater. Even though the Ravensburg plant is not quite completely developed, this new simple Escher Wyss model appears to be much more reliable than the St. Denis type. Several other plants of different power but based on the same simple design are being built by Escher Wyss.

Returning to the question of large complex plants, it can be stated that all those under construction or test in 1952 (20,000-kw and 15,000-kw English Electric, 15,000-kw Parsons, the St. Denis and Dundee Escher Wyss plants, the 20,000-kw Sulzer semiclosed plant and the 15,000-kw Westinghouse plant) have taken much longer to develop than was foreseen and in general the performance has not been satisfactory. One firm has permanently

[1] Numbers in brackets designate References at end of paper.

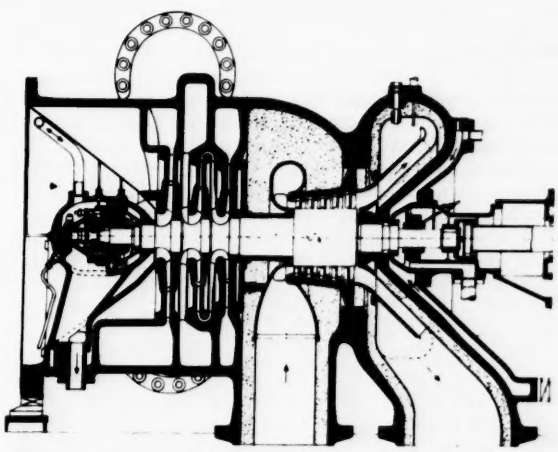

Fig. 2 Escher Wyss 2000-kw gas-turbine and compressor unit (Courtesy American Turbine Company)

abandoned the complex-cycle unit. The Sulzer semiclosed cycle has been abandoned by the manufacturers and the 20,000-kw unit installed in the Weinfelden, Switzerland, power station is being replaced by two simple open-cycle 10,000-kw sets. The development of the semiclosed-cycle Weinfelden unit was considerably delayed by trouble, but at the end gave excellent results as regards power output, efficiency, reliability, and control. Unfortunately, these results could only be obtained when running

with distillate oil, which was much too expensive for a base-load generating plant. Notwithstanding the low turbine-inlet temperature of 1200 F, excessive deposits in the power-turbine circuit formed very rapidly when this unit ran on residual oil. Sulzer undertook a very expensive and complete research program in the hope of finding a suitable fuel additive to prevent these deposits. Although this program gave very interesting results, it proved impossible to run the Weinfelden unit satisfactorily on residual oil. This was due mainly to the fact that, in the semiclosed cycle, the power-turbine circuit runs on a fuel-air mixture unusually rich for a gas turbine, and the expansion ratio in the circuit is very high, about 16 to 1.

Small industrial gas turbines and aircraft gas turbines have avoided some of the development troubles of the large industrial units. In the latter, it is impossible for practical reasons to test the main components separately. It must be realized that a 15,000-kw gas turbine has a turbine which develops about 45,000 kw while the compressor absorbs about 30,000 kw and the air flow through the combustion chamber is of the order of 300 lb per sec. No firm has the facilities for such tests, so the complete machine has to be run in order to try the components. In a small gas turbine, the components are of moderate size and power, and can be tested independently. Even though aircraft gas turbines develop very large powers, they are generally intended to be built in quantity, and suitable test facilities exist either at the manufacturer's or government testing stations such as NACA or the British National Gas Turbine Establishment.

AIRCRAFT GAS TURBINES

The tendency toward higher pressure ratios in jet engines, mentioned in the 1952 Report, has been confirmed. It is very difficult to obtain high pressure ratios (over 7 to 1) in a single axial compressor. Owing to the great difference in density between the l-p air at the inlet and the h-p air at the exit, the blades of the first stages are too long and those of the last stages too short. The twin-spool engine is the answer to this problem, Fig. 3. A h-p compressor section and a corresponding h-p turbine section, joined by a hollow shaft, are interposed between the l-p compressor and the l-p turbine which are connected by a thin solid shaft passing through the hollow shaft of the h-p section. In this way, each section can have the most appropriate blade length and rotational speed, and a very compact machine is obtained, with pressure ratios of 12 to 1 and more. The h-p and l-p parts of the machine are not mechanically connected, so that their speeds of rotation can and do vary under different operating conditions. This improves the matching of compressors and turbines and gives better part-load efficiency. Many modern jets, such as the Pratt and Whitney J-57 and J-75 and their civilian versions, and the Bristol Olympus, and turboprops such as the Rolls Royce Tyne are of the twin-spool type.

The high pressure ratios obtainable with twin-spool engines give a good specific fuel consumption and are suitable for commercial and military aircraft flying at high subsonic speeds. There are, however, several types of military aircraft designed to fly normally at supersonic speeds. The jet engine for these does not need a high compressor-pressure ratio because the ram effect obtained at supersonic speeds in the air intake provides a powerful supplementary compression. The use of a single low-pressure-ratio compressor with few stages results in considerable savings in weight and cost. The first engine based on this concept was the De Havilland Gyron, and several prototypes of other makes are on test. None of these engines is operational yet and their future depends on whether supersonic military aircraft will survive the onslaught of missiles. Bypass engines, although specifically designed to give very low fuel consumption at subsonic speeds, can also be used very successfully for supersonic

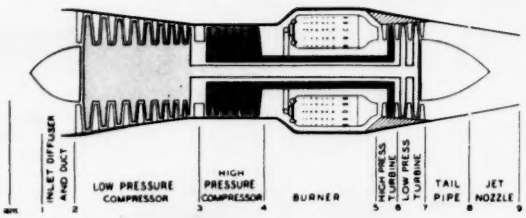

Fig. 3 Twin-spool turbojet engine (Courtesy Pratt & Whitney Aircraft)

aircraft. In this case also, the supersonic ram effect is used to increase the cycle-pressure ratio of a light and simple machine. This will be discussed further in the paper on aircraft applications.

GAS TURBINE COMPONENT PARTS

Compressors

Centrifugal

Centrifugal compressors are still mainly used in small industrial and automotive gas turbines and in very small aircraft engines, such as the Turboméca jets and turboprops. In large industrial machines, Escher Wyss is apparently the only firm still using centrifugals in the multistage, intercooled form, as shown in Fig. 2. Maschinenfabrik Oerlikon seem to have given up development of their gas turbines with the special centrifugal compressor but not of the compressors themselves for industrial use. British Thomson-Houston built two gas turbines with Oerlikon-type centrifugal compressors under license, but have discontinued this successful line because a change in engineering personnel brought to the fore an axial-compressor proponent.

In smaller gas turbines, the centrifugal holds its own even in the medium-power sizes. The new Ruston and Hornsby model TE 500-hp industrial gas turbine has a centrifugal compressor with moderate pressure ratio, Fig. 4. Almost all automotive gas turbines, in the United States and abroad, use single-stage centrifugals. Moderate-size industrial and auxiliary marine gas turbines use mostly centrifugals, but the Solar Jupiter and the larger Allen models use axials. The smallest axial in practical use is on the Solar Jupiter and has an air flow of about 8 lb per sec. This seems to be the lowest practical size in axial compressors. The pressure ratio obtained normally in a single centrifugal stage varies generally between 3 and 4. When higher ratios are wanted, two stages are often employed, as in the Fiat and Austin automotive prototypes, now obsolete. The old Rolls Royce Dart turboprop, illustrated in the 1952 Report, but still going strong as the most successful commercial-aircraft gas turbine, also uses a centrifugal compressor.

In industrial machines, the pressure ratios per stage are very low. Escher Wyss uses three stages to obtain 3.5 to 1 and British Thomson-Houston four stages to obtain 4.6 to 1, so that the average ratio per stage is about 1.5. Some firms, however, aim at high pressure ratios in single-stage centrifugals, in order to increase the cycle efficiency without the complication of heat exchangers. Both Boeing and AiResearch have experimental machines with compressors giving 6.5 to 1 in one stage. The adiabatic efficiency of these compressors already achieved is not bad and, as a result of further development work, efficiencies of 78 to 80 per cent will probably be obtained—quite high values considering the small size and low Reynolds number of these compressors, which have an air flow of under 5 lb per sec. These high pressure ratios are reached with ordinary compressors and radial-bladed open-forged aluminum impellers. The 6.5 ratio requires peripheral velocities of over 1700 fps and represents very

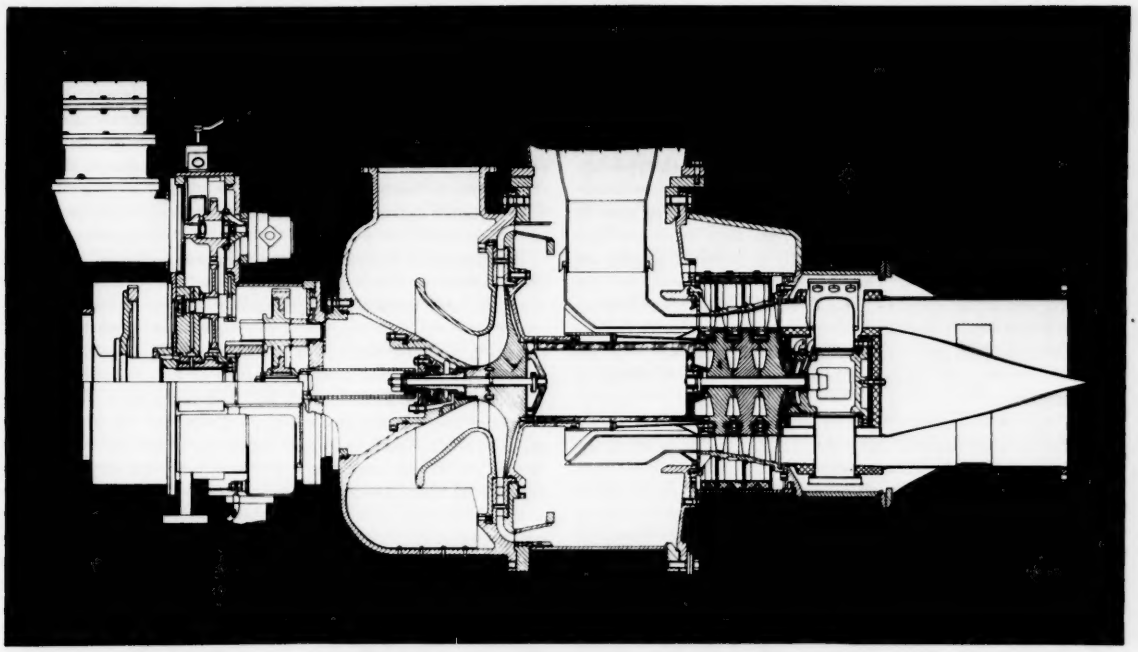

Fig. 4 500-hp model TE gas turbine (Courtesy Ruston & Hornsby Ltd.)

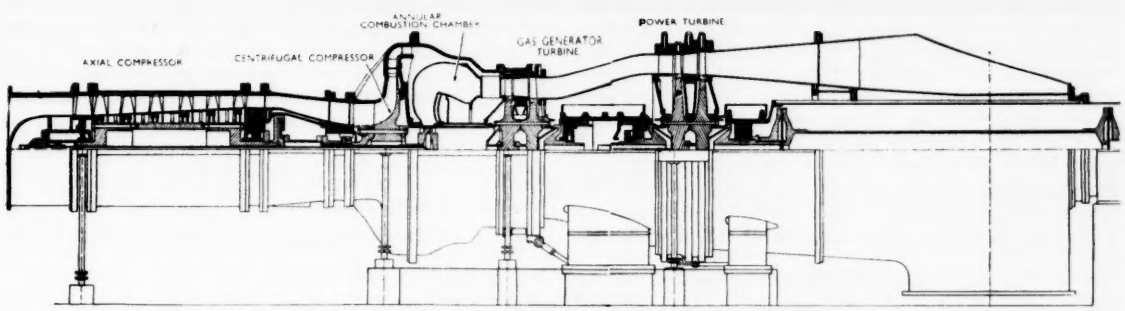

Fig. 5 Schneider-Creusot-Turboméca 6000-kw gas turbine (Courtesy *Oil Engine and Gas Turbine*)

nearly the maximum ratio obtainable with aluminum impellers, because the heating of the air is so considerable—over 500 F—that the stress resistance of the metal begins to be impaired. Were it not for this, even higher ratios could be obtained because well designed rotors stand peripheral speeds of over 2000 fps which would allow ratios of up to 10 to 1. In these compressors the aim of the development work is to obtain equally good efficiencies in each compressor component: Inducer, impeller, diffuser, and exit channels. Some interesting papers on this subject have been published by the Boeing engineers [2].

To improve the efficiency of centrifugal compressors, double-entry impellers and curved impeller blades have been tried with varying success. Double entry has the advantage of permitting the airflow to double without increasing ventilation losses. While the scrubbing losses are doubled, the loss on the back face of the impeller is, of course, eliminated. Another advantage is that there is no axial thrust The main difficulty resides in the fact that the air inlet at the front and at the back cannot be exactly symmetrical and it is consequently very difficult to avoid unbalanced flow on two sides, which causes unpleasant resonance phenomena. Of course, double-entry compressors have been used in the early centrifugal Rolls Royce and General Electric jets, now obsolete; but in these the ram effect of the entering air permitted a better adjustment of flow. While curved blades certainly improve impeller efficiency, particularly at the inducer end, they are unsuitable for high peripheral speeds because of the bending stresses on the blades. A reasonable compromise consists in having blades curved at the entry but ending radially at the exit. The Escher Wyss compressors use shrouded and curved blades, as in industrial blowers, but the peripheral speeds are low.

It is a pity that no firm has had the courage to try a large single-stage centrifugal of modern design for a large industrial gas turbine. It is probable that the high air flow and Reynolds number would allow very good efficiency to be obtained together with the reliability, simplicity, flat characteristic curves, and low cost which are the features of good centrifugals. However, when tests on the new Schneider-Creusot-Turboméca 6000-kw unit, which has a combined axial-centrifugal compressor, are complete and become available (see next section and Fig. 5), some useful indications on the performance of large modern centrifugals will be available. It must be noted that excellent results have been obtained on centrifugals built by large turbosupercharger manufacturers. For instance MAN, which started high-pressure supercharging for their very high efficiency large diesel engines with

axial compressors, now use centrifugals exclusively with much easier matching, lower costs, and adiabatic efficiencies of the same order as those obtained with axial compressors.

Combined Axial and Centrifugal

In the 1952 Report the Bristol Proteus turboprop and the 2700-hp English Electric gas turbine were described. Both types are continued, the Proteus powering the Bristol Britannia transatlantic airliner and the English Electric turbine being now in batch production; in both cases, however, the final centrifugal stage compresses little and has mainly the function of helping the air turn around the corner. The successor of the Proteus, the Bristol Orion, has a twin-spool axial compressor. The recent Lycoming small aircraft gas turbines (helicopter engines and turboprops) also have combined axial and centrifugal compressors. Here too the main function of the centrifugals seems to be the change in flow direction.

The Schneider-Creusot-Turboméca 6000-kw unit has a real combined axial-centrifugal compressor, Fig. 5. This is one of the five different 6000-kw prototypes which the nationalized Electricité de France ordered with praiseworthy public spirit to encourage the development of the industrial gas turbine. These units which normally are used as synchronous phase correctors are intended to drive auxiliary generating sets when necessary. High efficiency is not required for this duty, as only 300-hr-per-yr operation is foreseen; ease of starting, simplicity, and moderate cost are the main requirements. The unit was built by the big Schneider-Creusot armament works, but the design was entrusted to Turboméca. A 6-stage axial compressor, mounted on plain bearings, gives a pressure ratio of 2.5 to 1. It is connected through a flexible joint to an h-p shaft, comprising the single-stage centrifugal compressor (pressure ratio 1.8 to 1) and the two-stage compressor turbine. This h-p shaft is mounted on ball bearings, a most unusual solution for an industrial machine of this type. The transition from axial to centrifugal compressor seems to have been studied with great care. A diffuser of considerable length follows the last axial stage and a fixed row of guide vanes immediately precedes the inducer to the radial impeller. The diffuser and diffuser exit are of typical Turboméca-Szydlowski design and so is the annular combustion chamber with the characteristic rotating fuel nozzles. The 2-stage power turbine is mechanically independent of the compressor turbine, being mounted on plain bearings. It also has two axial stages of considerably greater diameter than those of the compressor turbine, to which they are joined by a long diffuser. Another long diffuser follows the power turbine before the exit volute is reached. The power turbine is connected to the generator by means of a long shaft mounted between two flexible couplings. A dog-clutch is fitted between turbine and generator so as to allow the generator to turn as phase corrector while the turbine is stationary. Several of the Electricité de France units have these couplings which can be connected and disconnected at speed but under no-load conditions, a most unusual and interesting feature considering the power involved. In the Schneider-Creusot-Turboméca machine, the compressor turbine runs at 5200 rpm and the power turbine at 3000 rpm; the cycle pressure ratio is 4.5 to 1, the turbine-inlet temperature 1300 F, and the air flow 132 lb per sec. The specific weight of the unit is only 6.4 lb per hp, very low for an industrial unit. The test results of this machine, which has by far the largest centrifugal compressor of any industrial gas turbine, will be followed with great interest.

Several studies and proposals have been made to connect a centrifugal compressor with one or two transonic axial stages to be described more fully under Axial Compressors. As each transonic stage can give pressure ratios up to 1.4 to 1, the combination of one stage with a 4 to 1 centrifugal would give a cycle-pressure ratio of nearly 6 to 1, and two transonic stages give about 8 to 1. At these high pressure ratios, small light gas turbines can give very favorable specific fuel consumptions, down to about 0.6 lb per hp-hr, without need for a heat exchanger. However, the combination of transonic axial stages and centrifugal impellers is not easy and up to now no successful test results have been published. The transition between axial and centrifugal stage is difficult as the radial-equilibrium conditions are essentially different for the two. Also, transonic stages work satisfactorily only very near design point and are difficult to match, so that the flat characteristic curve of centrifugals is lost.

Centrifugals With Rotating Diffusers

Although this type of compressor, described in the 1952 Report,

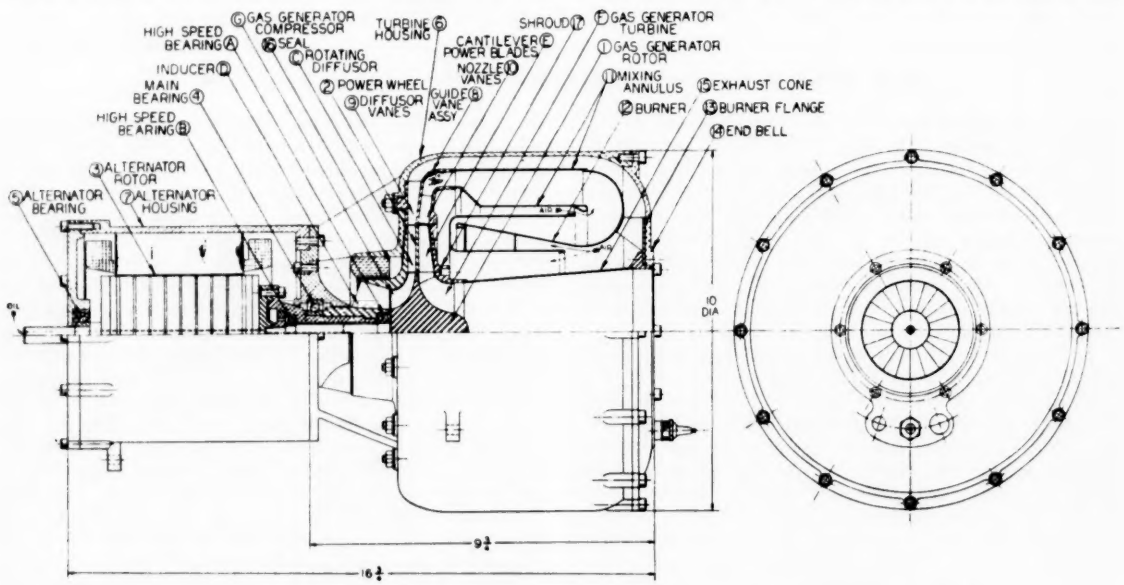

Fig. 6 7-kva Turbo-Mite gas turbine in cross section (Courtesy Propulsion Research Corporation Division of Curtiss-Wright Corporation)

has been abandoned by Saurer in favor of ordinary centrifugals, there has been a recent revival of interest. A very interesting application has been incorporated in the 7-kva Turbo-Mite[2] gas turbine which is almost surely the smallest gas turbine, Fig. 6. In this machine, which, notwithstanding its small size, has independent gas generators and power shafts, the rotating diffuser is used to add pressure to the air coming out of the impeller and not, as is usually the case, to take power out. The 3-in-diam gas-generator section, which turns at 90,000 rpm, consists of a mixed-flow centrifugal impeller and mixed-flow inflow turbine (Birmann wheel) cast back to back in a single piece. The inducer to the compressor, the radial rotary diffuser which follows the compressor, and the radial-inflow power turbine form a single piece with 6-in. max diam and turning at 24,000 rpm. This piece is cantilevered on the generator bearings, so that inducer, rotating diffuser, and power turbine are overhung. The high-speed-gas-generator shaft is mounted on bearings running in the power shaft, so that the rubbing velocity is the difference between the two speeds; also the combined high-speed compressor and turbine is overhung. A very simple annular combustion chamber completes the machine. The air flow is only 0.4 lb per sec and the cycle-pressure ratio is 2.5 to 1. The combined rotating diffuser and rotating-inflow power turbine are used to obtain a relatively low power-shaft speed of only 24,000 rpm notwithstanding the small diameter and air flow. In this way, a reduction gear is not required, and simplicity and efficiency are greatly enhanced. It must be noted that in this machine the power turbine is on the h-p side, while the compressor turbine is on the l-p side, an unusual arrangement. In this way the power-shaft speed is directly controlled by the combustion-chamber temperature, a useful feature in an AC generator which must keep constant frequency. It will also be noted that two inflow stages are used, no fixed stator being interposed between them. The high-speed turbine which runs at 1075 fps is the cold turbine. The rotating diffuser has a tip speed of only 573 fps.

Axial

The great majority of gas turbines still use axial compressors of the multistage type In general there are from 6 to 16 stages, according to pressure ratio desired, peripheral velocity, aerodynamic blade load, and type of blading diagram. It is difficult to obtain a pressure ratio much over 6 to 1 from a single axial compressor of the industrial type. With the highest speeds and greater blade lengths permitted in aircraft compressors, the ratio obtainable is higher, about 8 to 1. In aircraft engines, if even higher ratios are required, the twin-spool arrangement is used. In industrial turbines, two compressors with interposed intercooler are generally used, as in the very successful 22,000/30,000-kw Brown Boveri units with reheat. However, one industrial gas turbine, the Swedish STAL (Ljungström) uses a twin-spool arrangement, Fig. 7, similar to that employed in aircraft jet engines; but, of course, a separate 3-stage power turbine is used at the l-p end. The 10-stage l-p compressor is driven at 5500 rpm by a 2-stage medium-pressure turbine; the 10-stage h-p compressor is driven at 7000 rpm by a single-stage h-p axial turbine which is highly loaded so as to reduce the turbine-inlet temperature of 1382 F as quickly as possible. The cycle-pressure ratio is 10 to 1 and allows a thermal efficiency of over 24 per cent to be obtained without need for a heat exchanger. Multiple can-type combustors are used and the unit is very compact indeed.

Several U. S. gas-turbine prototypes, in the design or early development stage, use the new transonic axial compressors already mentioned. Some of these prototypes are jets, some are marine units, some are small machines for auxiliary military use. The transonic and supersonic compressor are not new in conception. Before the end of World War II, tests were made in the United States with highly supersonic stages and pressure ratios as high as 6 to 1 per stage were obtained; but the efficiency was very low and reasonably satisfactory operation could only be obtained at a given value of the airflow and the pressure ratio. The characteristic curve had shrunk to a single point. Soon after the war, very interesting tests were carried out in Europe with transonic stages employing thin profiles and operating at relative velocities near to the speed of sound, Mach 0.8 to Mach 1.2. The results were excellent, a pressure ratio of 2 being obtained with only two stages and about 86 per cent adiabatic efficiency. These European tests were carried out for military applications which did not materialize, and the results were never published. A similar development was successfully carried out in recent years by NACA; unfortunately the reports are still classified. The NACA results are similar to those obtained in Switzerland; pressure ratios of about 1.4 per stage can be reached at good stage efficiencies, but the characteristic curves are steep. Transonic compressors operate at Mach numbers between 0.8 and 1.2, they use very thin profiles and very high solidity (chord/pitch ratio), much higher than in ordinary axial compressors. The limits to blade loading are determined with criteria which differ from those used in ordinary compressor calculations. In the last few years, several

[2] This is presently under development by Propulsion Research Corp., subsidiary of Curtiss-Wright Corp.

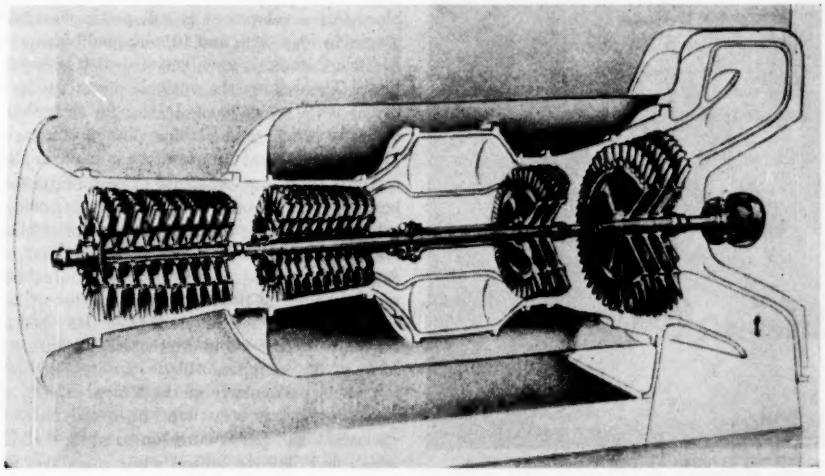

Fig. 7 STAL 10,000-kw twin-spool compressor gas turbine (Courtesy *Oil Engine and Gas Turbine*)

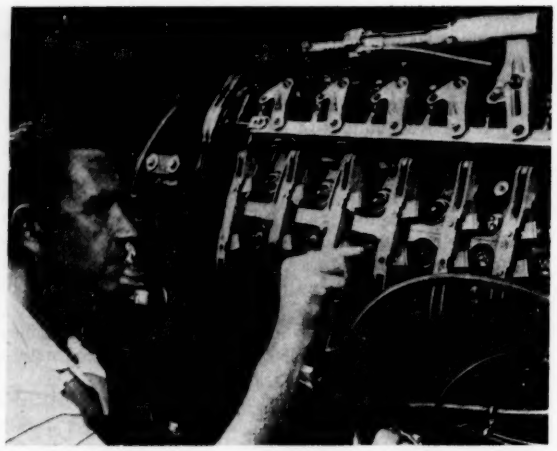

Fig. 8 Stator blades for J-79 jet-engine operating mechanism (Courtesy General Electric Company)

firms have attempted to build transonic compressors for prototype gas turbines, large and small, aircraft and nonaircraft. Apparently the task is not easy and only one firm seems to have been completely successful. Other attempts to use transonic stages, either as full axial compressors or in combination with a centrifugal impeller, have been given up. It is difficult to give a valid opinion on this development, there is not yet enough information. It is possible that the transonic compressor will continue to be limited to aircraft and special gas turbines where light weight and small diameter are of paramount importance. It must be noted that, even with ordinary axials, many industrial firms prefer to limit the load on the blades at the cost of using more stages so as to obtain better efficiency and flatter characteristic curves.

One way of obtaining better matching and flatter curves is to use controllable stator blades. These are similar in mechanism and operation to the adjustable turbine stator blades described in the 1952 Report and, like these, have been pioneered by General Electric. While the adjustable turbine blades are mainly used on split-shaft industrial machines with the purpose of adapting the velocity triangles to varying load and speed conditions, the adjustable compressor stator blades are used mainly on aircraft gas

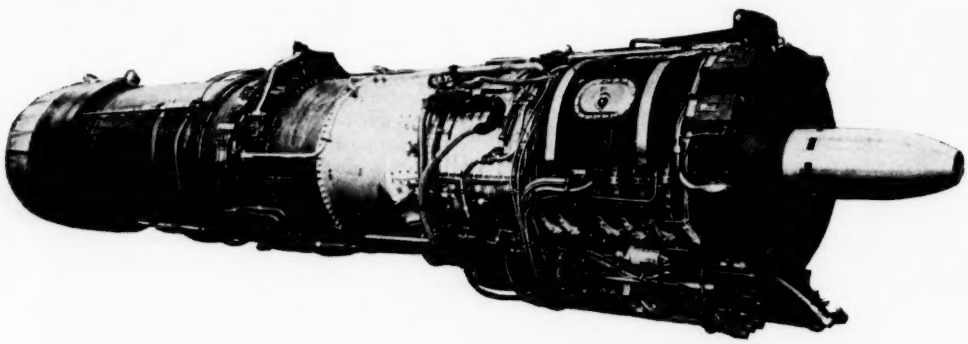

Fig. 9 J-79 jet from compressor side, stator blade adjustment is visible (Courtesy General Electric Company)

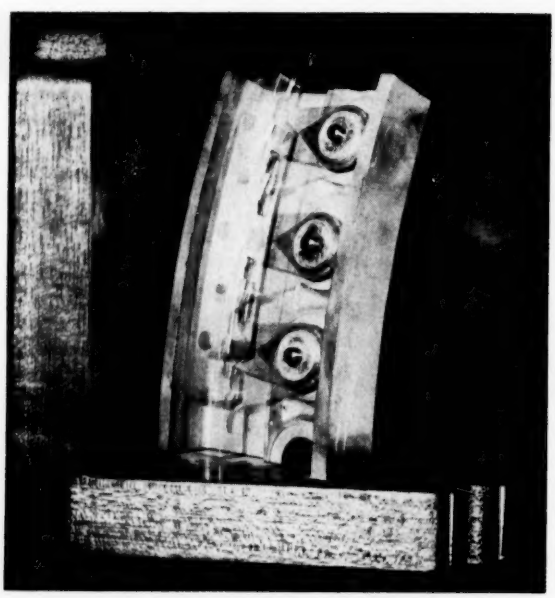

Fig. 10 Double-exposure photograph of J-79 stator blade (Courtesy General Electric Company)

turbines to help in matching the compressor to changing ram and altitude conditions. Tests have also been made by General Electric on an industrial gas turbine compressor mainly with the purpose of reducing air flow, and consequently fuel consumption, when idling [4]. These tests were carried out on the inlet guide vanes and the stators of the five l-p stages. For aircraft gas turbines this development is still partly classified, but the photographs in Figs. 8, 9, and 10, obtained by courtesy of the General Electric Company, have very recently been released for publication. These show the variable stator blades on the 10,000-lb-thrust J-79 jet engine. It can be seen that here too several stator rows are controllable. The double-exposure photograph, Fig. 10, shows that the variation in blade angle is considerable.

Apart from improvements in performance, considerable work has been done on axial-compressor-blade production and mechanical details. Owing to the great number of blades, a reduction in blade cost can affect notably the cost of the unit, and blade costs depend, of course, on the production method employed. The most economical production method for quantity-produced aircraft engines is very different from that used for large industrial machines where a hundred units would be a large quantity. Most of the compressor blades are in steel and there is a notable tendency, particularly in the United States, to use cast blades. Much work has been done on aerodynamically excited blade variations and particularly on rotating stall [5]. This problem, which was only recognized a few years ago, has since then been very thoroughly explained and analyzed. There are always

difficulties about blade vibrations, particularly in large machines. In many jet compressors nonfree-vortex blades often have substantially constant chord and thickness from hub to tip, and this makes the vibration problem even more complicated. On the whole, however, some progress has been made in this field.

Nuclear

The compressor type to be used in nuclear gas turbines is already under discussion. This problem is more urgent than would appear at first sight because large circulating compressors are also needed for nuclear steam plants when gas-cooled reactors are used. This is done in the many large British low-temperature gas-cooled plants already working or under construction; and the high-temperature gas-cooled reactor is rapidly gaining favor both in Europe and in the United States. There is a special compressor problem in nuclear plants because the coolant used in the reactor is not air but another gas, and the compressor design is strongly affected by the gas used. Except for special cases, such as aircraft applications, nuclear gas turbines will of course be of the closed-cycle type and use very effective heat exchangers. An effectiveness of 93 per cent is not unusual in nuclear projects. These highly recuperative cycles require relatively low cycle-pressure ratios which seldom exceed 3 to 1 and are usually down to 1.7 to 1 or less. Of the many organic gases and hydrocarbons that have been proposed as reactor coolants the choice seems to have narrowed to carbon dioxide and helium [6]. Carbon dioxide is cheap and plentiful, and works very satisfactorily at moderate temperatures. The British and French low-temperature reactors use it in their low-temperature gas-cooled steam plants, with centrifugal-circulation compressors. At high temperatures, such as are needed in a gas-turbine plant, carbon dioxide cannot be used directly over graphite moderators because it dissociates in the presence of carbon. Either a different moderator must be used or the graphite must be enclosed in stainless-steel containers and this does not make conditions in the reactor any easier. Also, recent experiments seem to prove that stainless steel is strongly attacked by high-temperature carbon dioxide in the strong neutron flux of the reactor. The specific heats of carbon dioxide are strongly variable with temperature. It has a moderate neutron-capture cross section and becomes radioactive in a reactor. Its molecular weight is high (44 against 29 for air and 4 for helium) and consequently has a low specific heat and a low velocity of sound, about 875 fps at atmospheric temperature. The number of stages required to reach a certain pressure ratio, i.e., a certain temperature rise, is proportional to the specific heat and inversely proportional to the square of the peripheral speed. Carbon dioxide proponents contend that very cheap compressors can be built for it as the low specific heat requires few stages. Helium proponents reply that this is only true if old-fashioned slow machinery is used. Carbon-dioxide compressors must run at very moderate tip speeds to maintain low Mach numbers with the low velocity of sound; consequently the number of stages is increased. Helium on the other hand has a very large specific heat but also a very high velocity of sound (well over 3200 fps at ambient temperature) so that in practice there is no Mach number limitation however great the tip speed; rpm and speed are only limited by mechanical stresses. With good modern design, notwithstanding the high specific heat, the number of stages can be quite reasonable. For a pressure ratio of 1.7 a satisfactory compressor can be designed with 12 axial stages or two centrifugal stages using available data; better results can be obtained when further research will permit higher load on axial bladings. As a nuclear gas, helium is almost perfect. It is both chemically inert and has zero neutron cross section so that it does not become radioactive even in the strongest neutron flux. Helium also has excellent heat-transfer properties and is for all practical purposes a perfect gas with constant specific heats. On the other hand, it is relatively expensive, and owing to its low density, it tends to leak, so that the sealing problem becomes even more difficult.

Special research will be needed for nuclear compressors, particularly of the axial type, which will be almost mandatory in large plants owing to the great flow volume. It is quite possible that axial compressors for carbon dioxide will give lower efficiencies than similar machines for air or lighter gases. This loss of efficiency has already been verified in practice in compressors for very heavy gases, which have been built in large quantities for uranium-separation plants. This effect is not well known and deserves to be studied. Complex molecules, with three or more atoms, have a finite "relaxation time." This means that some of the several degrees of freedom which characterize these complex molecules do not respond instantly to external stimuli. When subjected to a very short external action the gas starts to behave as if it were monatomic or biatomic and requires a finite time to respond like a multiatomic gas. This change is connected with an increase in entropy which results in a thermodynamic loss. The fact that a finite relaxation time exists was originally discovered in carbon dioxide and subsequently verified in heavier gases, while biatomic gases, like air, and monatomic gases are free from this characteristic. Considering an axial compressor with the usual axial velocity of about 450 fps and a blade-chord axial projection of about 1 in., it can be seen that each blade acts on the gas flow for less than 0.005 sec, after which the flow must change direction under the action of the next blade. This is a very short time indeed so that the relaxation-time effect can be felt and cause efficiency losses for carbon dioxide as it does for heavier gases. Research is also needed in blading diagrams, owing to the very different velocity of sound in carbon dioxide and in helium. In carbon dioxide the Mach numbers are high and, to prevent transonic and supersonic losses, the velocity triangles must preferably be of the 50-per-cent-reaction type, which reduces the relative velocity for a given peripheral velocity. In helium the Mach numbers are certainly very low, consequently velocity triangles having very high degrees of reaction are advisable, so as to allow the highest possible pressure rise per stage. Usually the degree of reaction will be greater than 1, so that the flow is accelerated in the stator and consequently the boundary-layer conditions in the whole machine are improved. Research on ways to obtain very high aerodynamic loads with helium has recently been carried out in Europe with the purpose of reducing the number of stages required.

In the next few years large sums will undoubtedly be spent both in the United States and abroad in nuclear compressor research and in subsidiary problems such as seals. The final results will not be known for a long time, particularly as regards the final choice of gas. Taking into account all relevant considerations, the author has little doubt that helium will prevail in closed-cycle nuclear gas turbines. It must be noted that the British, who have by far the greatest practical experience with carbon dioxide for low-temperature gas-cooled reactors, seem to have concentrated exclusively on helium for high-temperature gas-cooled reactors. It is quite likely that in a very short time a high-temperature gas-cooled steam plant using helium will be running in England. It also seems possible that future large nuclear plants using high-temperature gas-cooled reactors will be combined gas-turbine-steam-turbine cycles rather than pure gas turbine.

Turbines

Axial

The axial turbine is still by far the preferred type and is used exclusively in all except a few small units. Since 1952 improvements have mainly affected details of construction and materials rather than the general layout. In industrial machines, General

Electric continues to use only two turbine stages, both for single-shaft and split-shaft machines, notwithstanding the high turbine-inlet temperature, 1450 F to 1500 F. This practice is continued in the latest modified 1958 models, including the large 21,800-kw machine. Incidentally these three new models are all single-shaft machines, confirming the trend toward simplicity. In these General Electric turbines, the first stage is practically of the impulse type so as to accelerate the gas in the first-stage stator and reduce its temperature when it hits the rotor. In the General Electric split-shaft turbines the variable area second-stage stator nozzles, described in the 1952 Report, are being continued and have proved successful. There are about 80 units of this type in service Other manufacturers of industrial machines both in the United States (Westinghouse, Clark) and abroad (Brown Boveri, Ruston and Hornsby, English Electric, Metrovick, STAL, and others) always use from 4 to 7 or more turbine stages. Jet engines, which used to have a single turbine stage, now often have two or three. This is the result of the higher cycle-pressure ratios now used. Shrouding of rotor blades tends to increase. Shrouded blades give smaller leakage losses and consequently better turbine efficiency while allowing wider blade-tip clearances to be used. The fact that the shrouds of adjoining blades touch each other gives a certain amount of lateral support to the blades and helps to prevent vibrations from spreading. On the other hand, the blade centrifugal stresses are considerably increased, and so is the cost. Shrouding does not pay for very long blades, but it can be used even with very high tip speeds and temperatures, as in the Pratt and Whitney J-57 twin-spool jet turbine.

Turbine inlet temperatures tend to increase with the help of better materials and of blade cooling. The advantages to be gained both as regards efficiency and power output when the inlet temperature is raised are so great that a considerable development expenditure is justified. It will probably be spearheaded by the aircraft-gas-turbine makers. The fuel economy in turboprops and bypass engines, unlike that of pure jets, improves notably when the temperature is increased. But active development work in this direction is carried out also for industrial gas turbines, particularly in Europe. The aim is an inlet temperature of 2000/2200 F, at which the efficiency of the best Diesel engines is reached and units of over 50,000 kw become practical. Above 1600 F the blades must be cooled and different cooling methods can be used: Air, water, steam, liquid metals, and sweat cooling with air or other gases. For temperatures of 2000 F and over, cooling must be provided not only for rotor blades and disks, but also for stator blades and probably for combustion-chamber linings and inlet volutes. The more cooling is provided, the greater the quantity of heat subtracted from the cycle. Unless a way is found to recover at least part of this heat, the increased power and efficiency obtainable through a temperature rise is considerably offset. Recent studies seem to prove that, at least for large industrial units, where a certain amount of complication is permitted, this problem can be satisfactorily solved. Blade-cooling progress is covered in another section of this report.

Radial

The radial-inflow turbine has made slow but steady progress in the field of small gas turbines. Many manufacturers have started development work in this field and many prototypes are on test. Radial-inflow turbines are generally used in conjunction with single-stage centrifugal compressors. The turbine rotors can be precision-cast in one piece requiring little machining, a tempting perspective if quantity production is required. The big automotive firms have done much to reduce costs and improve the quality of these cast wheels. Not many turbine models with radial turbines have actually been put on the market. Among these is the **Stad**, a 250-hp British unit for industrial and auxiliary use made by a subsidiary of the Standard Motor Company Ltd., Figs. 11 and 12. It is a very simple machine. The centrifugal-compressor impeller and radial-inflow turbine rotor are mounted back to back, the driving-shaft flange being sandwiched between the two. An air space divides the backs of the two rotors and a stationary, thin sheet-metal separating diaphragm insures that compressor air bleeds to the back of the turbine rotor and then joins the main combustion-gas stream in the turbine rotor. This design is simple and neat, but the stationary diaphragm between the two rotors causes a double scrubbing loss which could be avoided either by a true back-to-back design or by a rotating diaphragm. Scrubbing losses constitute a large part of the turbomachinery losses, particularly in small radial machines. The two rotors are overhung on a shaft carried by two ball bearings which has the reduction-gear helical pinion at the other end. The first critical speed of the shaft is well below the turbine-idling speed. Twin combustion chambers, rather small in dimensions and located tangentially to the compressor, are carried by the main-compressor cover. This cover carries also the sheet-metal turbine volute and stator, of very light construction. The cycle-pressure ratio is 3 to 1, the speed 24,000 rpm, the turbine-inlet temperature 1430 F, and the best specific fuel consumption 1.28 lb per hp-hr. These small turbines are not efficient, but over-all dimensions, weight, and quoted price are low. It is intended to build these turbines in quantities and an expected life of 5000 hr is promised. Another interesting gas turbine using radial turbines is the little Turbo-Mite 7-kva unit already described, Fig. 6 [3]. In this machine both power turbine and compressor turbine are of the radial type, no intermediate stator being used between the two radial stages.

An interesting gas turbine for automotive use has been studied by C. A. Parsons and Company Ltd., and is shown in Fig. 13 [7]. In this regenerative unit a centrifugal compressor is used and the air goes through disk-type regenerators to an annular combustion chamber. From there it passes to a centripetal first stage (really a mixed-flow stage) which drives the power shaft and then, after a second stator, through a proper centripetal stage which drives the compressor. It will be seen that the h-p turbine stage is the useful power turbine while the l-p turbine stage is the compressor turbine. This unusual arrangement has the advantage of putting the power turbine directly under the control of the fuel injected in the combustion chamber, thus avoiding a portion of the lag usual in automotive gas turbines between throttle operation and power response. It seems unlikely that this design will be built, because the firm is mainly interested in large gas-turbine and steam-generating sets and in nuclear power plants.

There is no sign of any application of inflow turbines for larger units. There seems to be a preconceived idea that radial-inflow turbines are only suitable for small machines, although there is no real reason why this should be the case. Studies were made in Europe to use radial-inflow turbines with adjustable stator blades as power turbines for medium-large closed-cycle nuclear plants to obtain easy reversing. Some research work was even done to obtain data on efficiency in reverse, but no practical work has come out of these preliminary studies [8]. Even free-piston-engine marine plants use axial turbines with clumsy and inefficient reversing arrangements such as the shrouded reversing turbine used on the free-piston ship *William Patterson*. A variable stator inflow turbine would be the logical solution both for reversing and for adjusting the nozzle area to the number of gas generators in operation. The free-piston engine manufacturers would like radial-inflow turbines for this purpose but, as a matter of policy, they do not make the turbines themselves. Turbine manufacturers in both the United States and Europe have not shown enough initiative to offer efficient radial turbines for the powers required, from 1000 to 8000 hp. Basic research is needed in the radial-inflow turbine field. Up to now these have been designed,

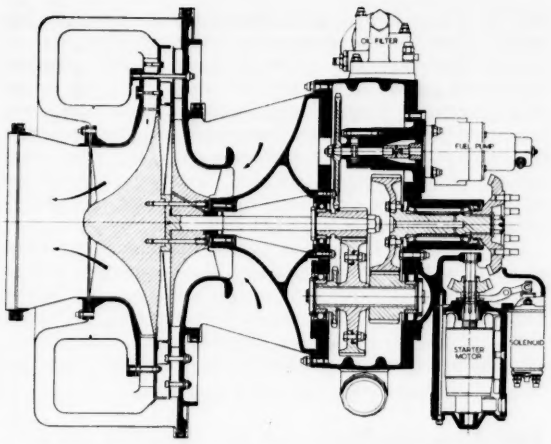

Fig. 11 Standard Motor Company Ltd., 250-hp gas turbine in cross section (Courtesy Oil Engine and Gas Turbine)

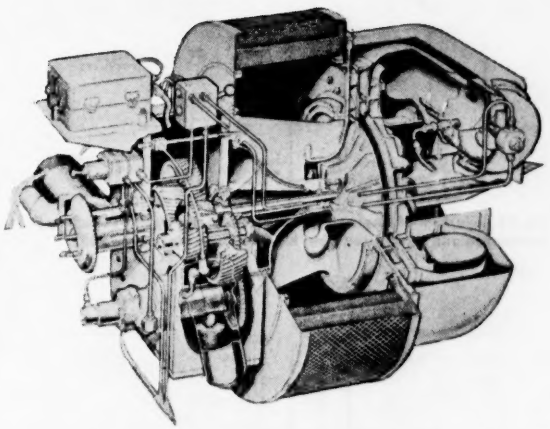

Fig. 12 Standard Motor Company Ltd., 250-hp gas turbine (Courtesy Oil Engine and Gas Turbine)

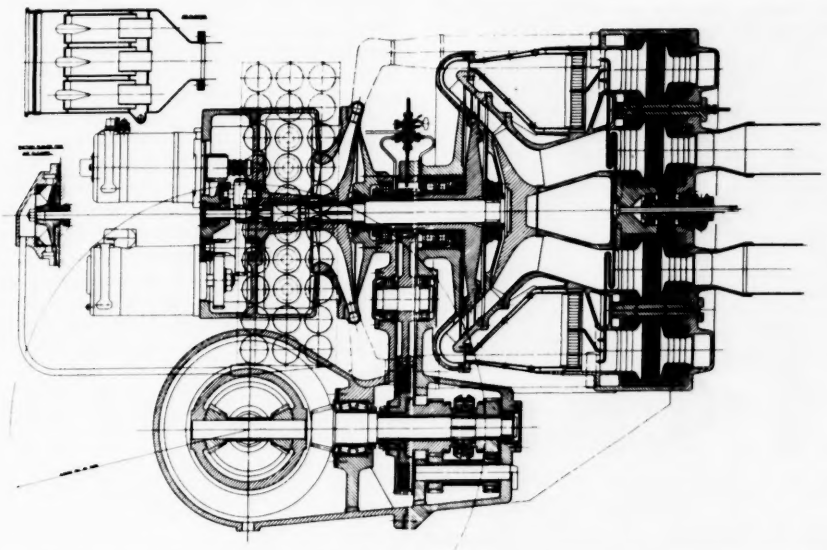

Fig. 13 Automotive gas turbine, showing two-stage inflow turbine (Courtesy C. A. Parsons Company Ltd.)

built, and tested, generally with good results, without much concern for the influence of various design parameters on performance, efficiency, and size. Fortunately, a comprehensive research program in this field is now being set up.

Turbine-Compressor Connection

In some of the applications of the gas turbine, particularly to road and rail traction, it is very important to obtain good torque characteristics at low speed and at no speed. The ordinary single-shaft gas turbine is, of course, useless for this purpose; when the power shaft slows down so does the compressor, mass flow and pressure ratio fall, and the machine stalls. The split-shaft gas turbine, however, in which the useful power is given by a turbine not connected mechanically with the compressor turbine, gives quite good torque-speed characteristics. Starting torques of about three times the design torque value and even more can be obtained, so that the need for a multispeed gear box disappears or is greatly reduced. But even the split-shaft turbine has unsatisfactory features. There is very little braking torque on the overrun; the response of the power turbine from an idling position is sluggish and the slow-running and idling fuel consumption is high. In recent years it has been proposed to connect compressor, turbine, and output shaft through a differential gear. This arrangement is called a differential gas turbine. Hutchinson has studied the use of a simple turbine connected to the compressor and to the power shaft through a differential gear, Fig. 14, and also that of a split-shaft turbine in which part of the power needed by the compressor is provided by a high-pressure turbine directly connected with it, while the rest is provided by the l-p turbine through a differential gear which also drives the

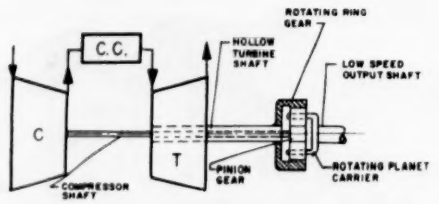

Fig. 14 Diagram of single-turbine differential gas turbine (Courtesy D. W. Hutchinson and reproduced from his SAE Paper No. 618, 1955)

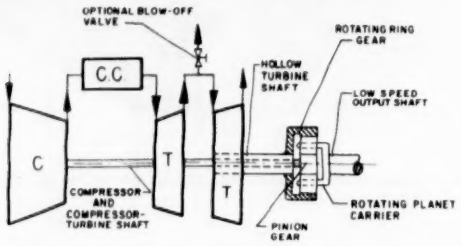

Fig. 15 Diagram of 2-turbine differential gas turbine (Courtesy D. W. Hutchinson and reproduced from his SAE Paper No. 618, 1955)

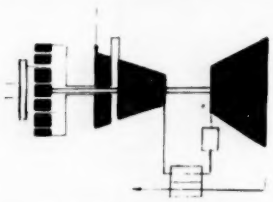

Fig. 16 Split-compressor differential gas turbine, l-p compressor, driven by differential (Courtesy C. A. Parsons Company Ltd.)

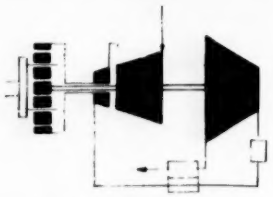

Fig. 17 Split-compressor differential gas turbine, h-p compressor driven by differential (Courtesy C. A. Parsons Company Ltd.)

power shaft, Fig. 15 [9]. The operating conditions are very interesting. Both the simple and the split-shaft turbine give good torque curves and idling and slow-running fuel consumption are greatly improved. If a blowoff valve is provided between the two turbines of the split-shaft unit and a brake is fitted on the power-turbine shaft, reverse torque can be automatically obtained through the differential. Many firms, particularly in the automotive field, showed interest in the differential gas turbine but no actual machine has been built, although studies are being continued. The fact that no actual market for traction gas turbines exists has not encouraged practical tests. The design difficulties and cost of the differential gear, which must be able to transmit much more power under some operating conditions than the useful power of the machine, have been additional deterrents.

Another type of differential gas turbine has been proposed by Bowden and Hryniszak of C. A. Parsons Company Ltd. [7]. A single turbine is used. The compressor is split in two parts, one driven directly by the turbine, the other connected to the power haft and the turbine through a differential gear. Either the h-p or the l-p part of the compressor can be directly driven, Figs. 16 and 17. The results that can be obtained have been examined in a recent paper and are very interesting. The high torque multiplication can attain values of over 4 to 1; and a considerable amount of braking torque has been predicted. The mechanical connection provides a quick response to load changes, and idling and slow-running economy are good. The main difficulty lies in matching the two compressors which must operate within an extremely wide range of speed and flow.

It is interesting to note that a differential turbine of another type can be obtained by using a centrifugal compressor with rotating diffuser. The turbine would drive the impeller directly and a differential gear would connect the power shaft, the turbine shaft, and the rotating diffuser. Time alone will show whether any differential turbine arrangement can be justified in practical use.

Combustion Chambers

In the last few years much work has been done on combustion and allied problems, from basic theoretical studies to experimental work on flame-holders and the effect of droplet size. The results have percolated slowly into actual combustion-chamber design. This has remained essentially empirical. There is less scientific foundation in a combustion-chamber layout than in any other gas-turbine component. In aircraft engines the annular type has gained a little on the canular and the multiple can. In this field the difficulties are greatest because frontal area and weight requirements are very stringent while high-altitude operation, particularly in military engines, introduces serious problems.

In large industrial gas turbines European firms in general continue to favor the single large combustor (Brown Boveri, Ruston and Hornsby) while U. S. firms prefer the multiple-can type. The new General Electric single-shaft units show, however, a change in the traditional General Electric layout [10]. In the older models the multiple-can combustors were directly interposed between compressor and turbine, as in a jet engine, thus lengthening the machine and the compressor shaft; even in regenerative units where a direct flow-through could not be obtained. In the new nonrecuperative models (Fig. 18 shows the largest type, a 21,800-kw single-shaft machine) the burner ends of the cans are located over the h-p compressor casing, thus considerably shortening the unit; the flow of the primary air is reversed. In the new General Electric recuperative units the eight cans are no longer placed symmetrically around the machine but are mounted in two "pods" of four cans each. The length of the machine is not reduced as much as it might be, probably because the same main castings are used as in the simple-cycle machines. The combustion of heavy oils and the fact that many gas-burning machines must be able to operate on oil under emergency circumstances, brings some complications to burner and combustor design.

In small gas turbines, the location and design of the combustion

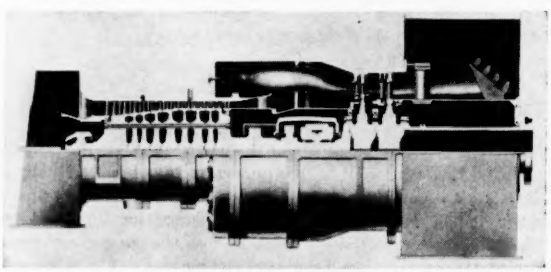

Fig. 18 GE single-shaft 21,000-kw gas turbine (Courtesy General Electric Company)

chamber are relatively more difficult. Space is generally at a premium and the bulk of the combustors and their piping is much greater in proportion to the size of the machine. The simplest solution is an annular chamber and Turboméca continues its rotating-spray chamber in the many successful models it offers. Without the rotating spray the short length allowed an annular chamber in a small machine makes complete combustion difficult. Twin combustors are often used, particularly with centrifugal-compressor machines. These allow a more symmetrical arrangement of the compressor exits; but getting both combustors to work equally in parallel is not easy and any unbalance gives rise to noise and vibration. In very compact small gas turbines, particularly those for automotive use, there is always the temptation to skimp on combustion-chamber length to gain room. Fortunately in most small-gas-turbine designs the combustion chamber can be altered and even radically changed without affecting the main components. Thus the most suitable type can be chosen by cut and try methods even after the units are in batch production.

Coal-fired air heaters of the radiant type have given good results and several new coal-fired Escher Wyss closed-cycle plants are being built or are on order. The 2000-kw Escher Wyss coal-fired plant at Ravensburg has been working satisfactorily for over a year; but the air heater is colossal in relation to the rest of the unit [1]. The total height is over 54 ft and the diameter of the vertical tube bundle is over 10 ft, Fig. 19. It is possible that, when more experience is gained with these coal-fired air heaters, their size may be reduced. From the economic point of view, the ability to burn a cheap and abundant fuel in a plant giving good efficiencies (25 to 28 per cent) even for moderate powers is of great importance; the more so as large quantities of hot water, which can be used for heating or industrial processes, are obtained free from the precooler.

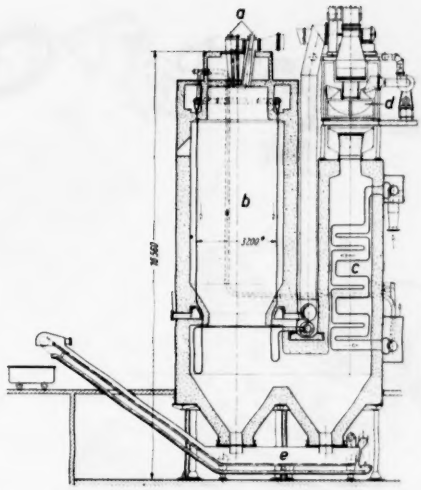

Fig. 19 Escher Wyss coal-fired air heater (Courtesy American Turbine Company)

Heat Exchangers

Although the trend is toward simplicity and away from recuperation, many industrial and marine gas turbines use heat exchangers. Most of the larger units, where cost matters more than weight, still use tubular heat exchangers. In open-cycle machines the effectiveness tends to be moderate, from 60 to 75 per cent. In closed-cycle plants a high degree of recuperation is essential and Escher Wyss has gone over to extended-surface tubes in the Ravensburg plant [1]. These consist of tubes with a continuous very thin steel band, 0.2 in. wide, wound helically in zig-zags by a special machine forming a continuous helical fin. The pitch of the fins is such that each row is offset relative to its neighbors, Fig. 20. The main tube is about 1 in. in diam and 0.040 in. thick. The h-p compressor air flows inside this tube and the outside fins lie in the l-p turbine-exhaust air. The inner tube, about 0.35 in. in diam, acts only as a support for the inner fins and its inside section is stopped. When the fins are being wound, a thin brazing-material foil is placed between tube and fins; the two tubes with their fins are assembled inside each other, the inner tube is then slightly extruded to insure contact between the inner fins and the inside of the outer tube, and then the whole lot is brazed together in a single operation inside a special oven. The fin portion of the tubes is 8 ft 4 in. long. The fin tubes are joined in bundles, in the geometric pattern shown in Fig. 20, and the triangular empty spaces between the outer fins are filled by triangular aluminum spacer rods. There are 1302 tubes in all. The counterflow temperature drop of the heat exchanger should be 54 F giving an effectiveness of over 91 per cent; in practice, however, the effectiveness has been lower, apparently because the brazing was not perfect and consequently the conduction heat transfer between fins and tubes was impaired. In the nuclear closed-cycle gas turbines now being studied, counterflow temperature drops of 48 F and about 93 per cent effectiveness are required. The excellent heat-exchange properties of the coolant gases, and particularly of helium, at the high pressures used in these plants (600 to 1000 psia) allow these high values to be attained with apparatus of moderate size.

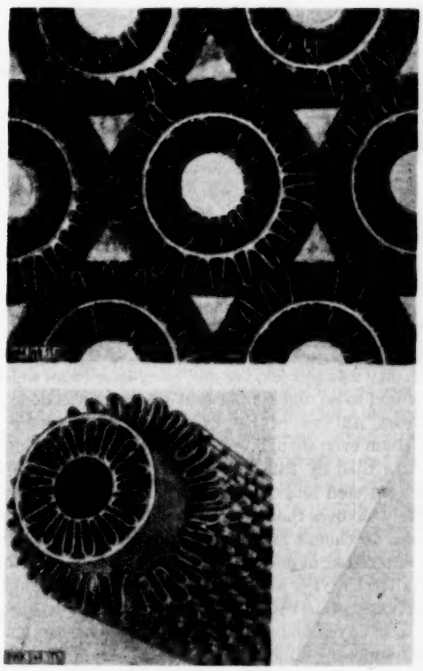

Fig. 20 Escher Wyss fin-tube heat exchanger (Courtesy American Turbine Company)

Small gas turbines for auxiliary and military use rely mostly on higher cycle-pressure ratios and better turbomachine efficiency to obtain reasonable fuel economy, without using heat exchangers. In the latest simple-cycle machines, a fuel consumption of about 0.7 lb per hp-hr is obtained with great simplicity and

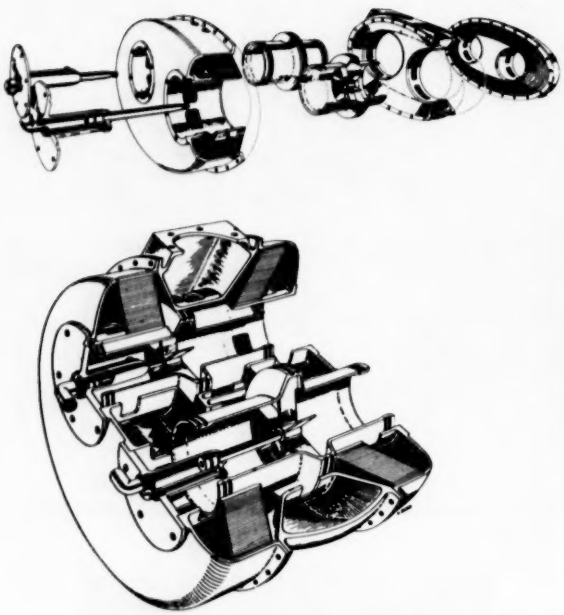

Fig. 21 Ritz Parson piston-type heat exchanger (Courtesy C. A. Parsons Company Ltd., and W. T. von der Nuell)

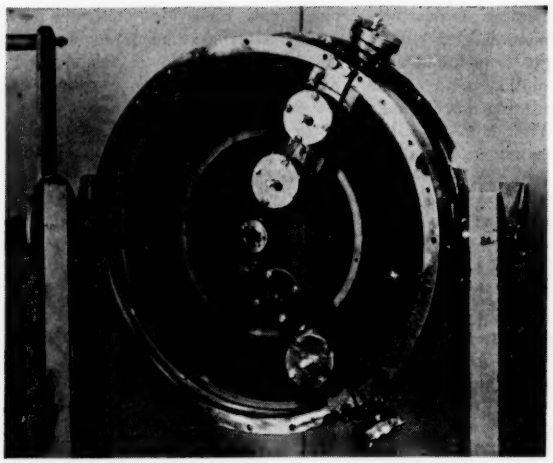

Fig. 22 Ford disk-type regenerative heat exchanger (Courtesy Ford Motor Company)

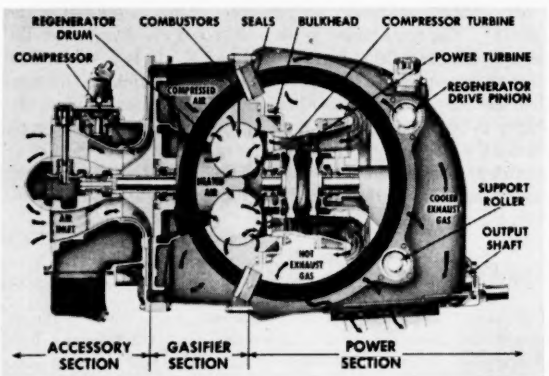

Fig. 23 GM drum-type regenerative heat exchanger (Courtesy General Motors Research)

extremely light weight. But the automotive gas turbine, particularly the passenger-car gas turbine, seems definitely to need the recuperative cycle to compete with a piston engine. In the last six years, a great amount of effort and money has been spent to develop automotive heat exchangers, with rather interesting results. The work on rotary regenerative heat-exchangers, described in the 1952 Report, was pursued in England not only by NGTE but also by Parsons who were trying to develop a regenerative gas turbine for British army tanks. The results obtained in England did not bear out the early promises. The main difficulty was seal leakage and carry-over losses which stubbornly refused to go under 4 per cent whatever was done to seals and matrices. Interest in the regenerative type waned considerably in Britain even though work was continued. A compromise solution was tried by Ritz at Parsons. Two fixed regenerative matrices were used and compressor air and exhaust gases were alternatively led over them by piston valves, Fig. 21. The situation seemed to change radically around 1955 when Detroit got seriously interested in gas turbines for passenger cars, following the lead of the Chrysler group. A Plymouth chassis propelled by a Chrysler gas turbine with disk-type regenerative heat exchanger was said to have reached a mileage per gallon comparable to that of the contemporary standard piston-engine Plymouth with automatic transmission. Ford also studied this type of heat exchangers and the published results claim very small leakage losses, Fig. 22 [11]. General Motors, which had originally begun with nonregenerative automotive units, followed suit with a drum-type regenerator, Fig. 23 [12]. Following these developments the automotive gas turbine seemed to be just around the corner and the rotary regenerator the perfect solution. Engineers were making rash claims and the manufacturers made even more rash promises of wonders to come.

Why was the regenerative heat exchanger so good in Detroit and not so good in England? The answer lies not in the regenerators themselves but in the difference of automotive developments in America and in Europe. Owing to the low cost of gasoline, American cars have never been particularly economical in fuel. Automatic transmission and the horsepower race have made them even more thirsty in late years, making competition with standard cars on the basis of miles per gallon progressively easier for the gas turbine. Moreover, the absurd horsepower race caused American cars to operate most of the time at a small fraction of the nominal power. While a European car would have a 50-hp piston engine or a 90-hp gas turbine, an American popular car has a 250-hp engine or gas turbine; and yet neither of them uses more than 40 hp in normal operation. Relatively speaking, a regenerative gas turbine is much more economical when operating at low part-load than a piston engine, in which the mechanical friction losses become proportionally heavier when the load falls. In a split-shaft gas turbine, mechanical losses are negligible and at part-load the compressor turbine runs slowly, reducing the mass flow, and even more the cycle-pressure ratio. But, in a strongly regenerative cycle, the thermal efficiency increases when the pressure ratio falls, the effectiveness of the regenerator rises when the mass flow decreases and, what is even more important, leakage losses fall rapidly, because both seal and carry-over leakages are proportional to the pressure difference between compressor air and turbine exhaust. These are the reasons which made the Detroit gas-turbine cars look so promising. Even though a regenerative heat exchanger does not help the engine much at full power, when the pressure ratio is 3.5 to 1 and the

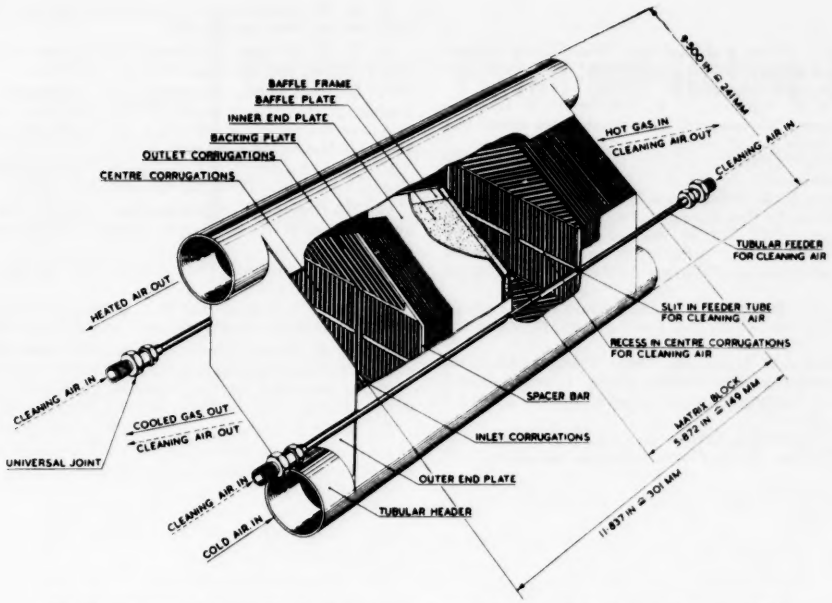

Fig. 24 Hexagonal recuperative heat-exchanger element (Courtesy C. A. Parsons Company Ltd.)

leakage 4 per cent, it is quite effective under quarter load with pressure ratio under 2 and mass flow and leakage greatly reduced.

The regenerative heat exchanger, unfortunately, has other defects. It is costly, difficult to fit into a compact configuration, the ducting is awkward and complicated, it needs a power drive, and its total weight is out of proportion with the weight of the bar matrix. The seals require maintenance and exhaust products and air dust foul the regenerator matrix. Although it is difficult to obtain reliable information on the latest developments, it is known that Ford has given up the regenerative heat exchanger and it seems probable that other firms will follow suit. Both in America and in England hopes are now concentrated on recuperative heat exchangers of the laminar type in which air and gas pass through grooves of very small hydraulic radius provided by standardized plates which can be manufactured cheaply in quantities and assembled in compact passages. Rumor has it that at least two of the Detroit auto makers and three British firms are working on laminar recuperators. Little published information is available on the subject except for the papers by Hryniszak [13, 14] and that by Bowden and Hryniszak [7], already mentioned, which give information on the Parsons developments. These include the regenerative-rotary and piston-type heat exchangers as well as the more recent recuperative type. This is composed of matrix blocks formed by 100 to 200 cells, each formed by one backing plate, two space bars, one inlet, one center, and one outlet corrugation. All parts forming a matrix are joined together by brazing, Fig. 24. The cells are small, about 1 by 4 in. It is interesting to note that Parsons, with their long experience on regenerative heat exchangers, now seem to favor the recuperative type for automotive use. Among the advantages claimed for it is a low parasitic weight, i.e., a high ratio between matrix weight and weight of the complete heat exchanger including ducts. The recuperative heat exchanger is also cheaper, easier to fit in the general layout, and requires no power drive. The hydraulic radius of the Parsons recuperator corrugations is very small, about 1 mm, with provision for periodical cleaning by blowing air and detergents through the cells.

An interesting form of heat exchanger for medium and large gas turbines is the steam-producing heat exchanger. The utilization of exhaust heat for hot-water and process-steam production is old. General Electric has heated feedwater, and Oerlikon and Ruston and Hornsby have used exhaust-fed boilers in well-known installations to produce process steam. In these cases, however, the steam was generated in real boilers, large and heavy, where subsidiary oil heating could be used to produce larger quantities of steam [15]. Recently, the possibility of using a heat exchanger in a gas-turbine exhaust to produce steam for driving a steam turbine has been explored [16]. If condensing water is available and the plant is large enough and operates for long periods, it can easily be seen that this arrangement can be more advantageous in some cases than a recuperator. An open-cycle industrial gas turbine, operating at modern cycle conditions, gives a little less than 20 per cent efficiency without a heat exchanger. If a moderately effective recuperator is used, the efficiency goes up to about 26 per cent but the power output falls because of the increased pressure loss. If the same size of recuperator is placed in the turbine exhaust to produce steam to drive a steam turbine, this can give nearly as much power again as the gas turbine, while the pressure losses in the recuperator are halved and the fuel consumption remains about the same. It must be noted that the steam-generating heat exchanger requires, if anything, less heat-transfer surface than the ordinary gas-turbine recuperator, if the effectiveness remains the same. This follows because the heat-transfer coefficient between walls and steam is much higher than that between walls and compressor air. The steam-generating heat exchanger becomes much more advantageous in closed-cycle gas turbines, particularly of the nuclear type. The very low cycle-pressure ratios, high heat-exchange effectiveness, and high coolant-gas pressures all permit the amount of work obtainable from the steam turbine to become considerably larger. Recent studies [17] show that, if cycle conditions are kept constant in a nuclear closed-cycle plant, the power output becomes more than three times greater and the cycle efficiency increases about 15 per cent when the heat exchanger is used for steam generation [18].

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18 The *Oil Engine and Gas Turbine* has been a rich source of information and illustrations for all this section of the Report.

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Compound Piston-and-Turbine Engines

The prime-mover combination of piston-and-cylinder components with a high-speed exhaust-gas turbine and, in some cases, a centrifugal or axial-flow compressor has assumed a variety of forms. Progress in the development of some of the more notable members of this family of compound engines is presented.

THE hybrid fruits of the marriage of the piston-cylinder reciprocating engine with the high-speed turbine have assumed a variety of forms. The most important ones are described schematically in Fig. 1, where the salient shaft-work features of each member of the family are specified.

The first and oldest member is the turbocharged engine which has already achieved technical success and wide commercial acceptance. Turbocharged aircraft engines of the spark-ignition type and 4-stroke-cycle diesels have been commercially accepted for 20 years. Today the turbocharged diesel is the standard engine for marine, tractor, and heavy-duty-truck applications. The only really new feature in this engine development is the advent of centrifugal-compressor and turbine components of high enough efficiency so that 2-stroke as well as 4-stroke diesels can now be successfully turbocharged. Previously this was not possible because of the higher scavenge-air requirement, the associated lower turbine-inlet temperature, and finally, the necessity of having a turbine-inlet pressure always of the order of 10 per cent lower than the supercharger discharger. Under these circumstances, only when both turbine and compressor efficiencies are quite high, is the shaft output of the turbine sufficient to drive the compressor.

The main purpose of this review is to cover the three younger members of the family pictured in Fig. 1, namely:

1 The *turbocompound engine*, as characterized by the Wright spark-ignition engine which powers the DC-7 and Constellation airliners. The Napier Nomad and Napier Deltic are compression-ignition engines of this class. All the mechanical-power production from the piston-engine and turbine components and also the power requirements of the compressor, and the load tie into the crankshaft.

2 The *crank-piston-and-turbine engine* with turbo-power output, as characterized by the recently concluded Orion engine development. Here all the piston-engine output is absorbed by the centrifugal compressor after transmission by the crankshaft and gearing. The net output comes solely from the free-power turbine.

3 The *free-piston-and-turbine engine* with turbo-power output as characterized by the French SIGMA Model GS-34 unit as well as several automotive-type engines under development in this country. From a thermodynamic point of view this engine is identical to the previous member of the family as all the piston-engine ouput is absorbed in a compressor, of the reciprocating-piston type, and the net shaft output to the load is derived only from the power turbine. However, with a piston compressor instead of a centrifugal (or axial) type there is no need for a crank to convert reciprocating work to rotary work.

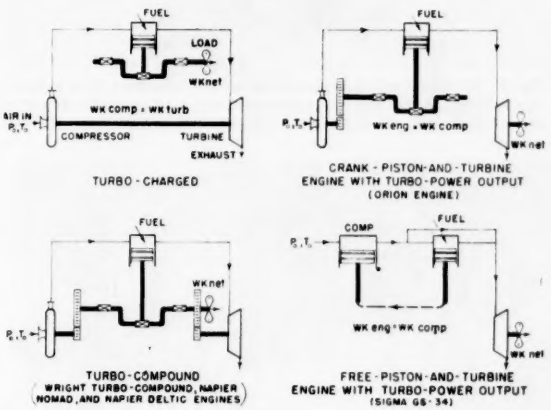


Fig. 1 The family of compound piston-and-turbine engines

After a primarily descriptive presentation of the existing engines in each class, some of the demonstrated advantages will be indicated along with some of the major problems encountered. An exhaustive treatment will not be attempted in this last respect. With due apologies to crank-engine partisans, somewhat greater emphasis will be given to the free-piston-type engine.

The Turbocompound Engine

As specified in Fig. 1, work terms produced by the piston engine and the exhaust-gas turbine and the work required for the supercharger are all fed into or are derived from the crankshaft for the turbocompound engine. This situation contrasts with the turbocharged engine where the turbine output runs the supercharger directly without mechanical coupling of these components to the crankshaft.

The Wright Turbocompound engine, Fig. 2, developed in the 1942-50 period, is the only commercial engine in this class. Three exhaust-gas-driven turbines are coupled to the crankshaft through reduction gearing and fluid couplings.

This engine, by deriving a substantial amount of power from the turbine components (15-20 per cent), has a 10 per cent lower specific weight and a 20 per cent better cruise specific fuel consumption than the conventional aircraft piston engines of the mechanically charged and turbocharged types. These are truly marked gains realized by a *revolutionary* engine design concept in contrast to the inching progress of the previous 20 years of *evolutionary* development of the aircraft piston engine.

The *Napier Nomad* engine, developed in England, is another example of the turbocompound class, Fig. 3. It is a 12-cyl,

Contributed by the Gas Turbine Power Division and presented at the Annual Meeting, New York, N. Y., November 30–December 5, 1958, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, July 9, 1958. Paper No. 58—A-46F.

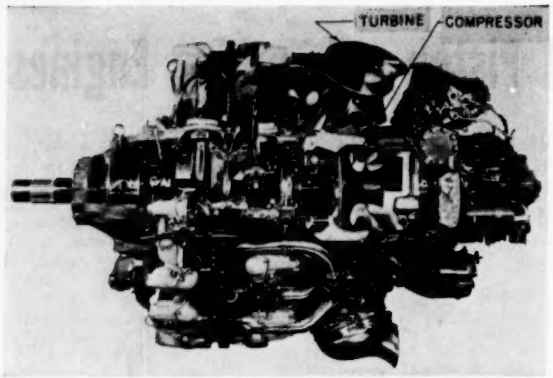


Fig. 2 The Wright turbocompound engine: Spark-ignition, three exhaust-gas turbines, 18 cyl, 3350-cu-in piston displacement, take-off rating 3250 shp at 2900 rpm, 265-psi mep at take-off, and 177-psi mep at cruise power

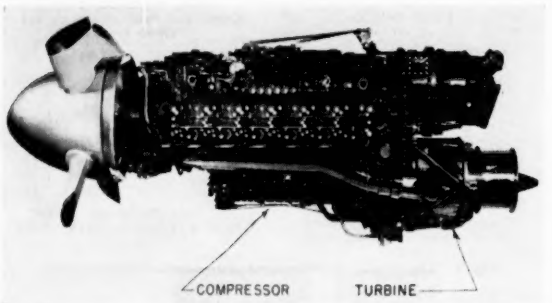
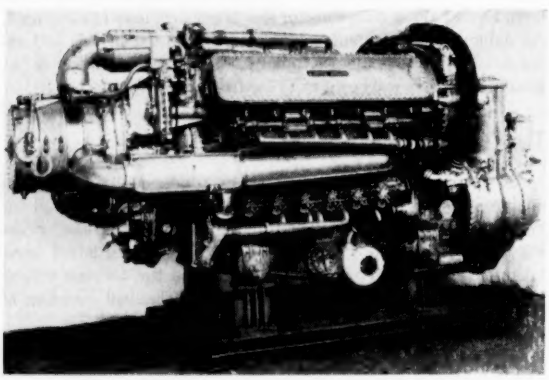


Fig. 3 The British Napier Nomad Engine: Compression-igniton, 12-cyl, 2-stroke loop-scavenged, 3100 hp, with 12-stage axial-flow compressor and a 3-stage turbine, displacement 2505 cu in, 2000 rpm; shaft powers: 2250 hp for turbine, 1840 hp for the compressor, 2660 hp for the engine at supercharge pressure ratio of 8:1



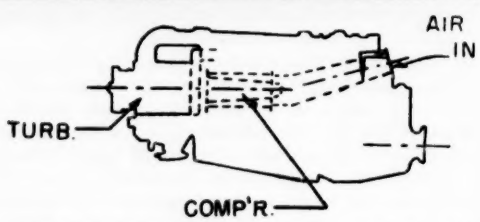


Fig. 4 The British Napier Deltic Engine: Compression-ignition, 18-cyl 2-stroke opposed pistons; axial-flow compressor and turbine; 5500 shp at 2000 rpm

2-stroke, loop-scavenged 3100-shp diesel engine with a mechanically coupled axial-flow compressor and turbine. It was designed specifically as an air-freighter power plant. A specific fuel consumption of better than 0.35 lb per hp hr has been claimed for it in contrast to the 0.40 specific for the Wright turbocompound.

The third example is the *Napier Deltic* engine, an 18-cyl, 2-stroke, uniflow-scavenged, 5500-shp diesel designed for locomotive and marine installations, Fig. 4. In contrast to the single-crank pancake layout of the Nomad, the Deltic is an opposed-piston layout with three crank shafts and the 18 cylinders in three banks of six each forming the sides of an inverted delta. Thus there are a total of 36 pistons in this remarkable engine. As in the case of the Nomad, the Deltic engine has an axial-flow compressor and a turbine mechanically coupled to the crankshaft. It has a specific weight of 2.5 lb per shp in contrast to 20 lb per shp for a conventional locomotive or naval marine-type diesel.

The Crank-Piston-and-Turbine Engine With Turbo-Power Output

The unique aspect of this engine relative to the turbocharged and turbocompound classes is that *all* the piston-engine shaft output is absorbed by the supercharger and the load is driven solely by the free-power turbine as emphasized in Fig. 1.

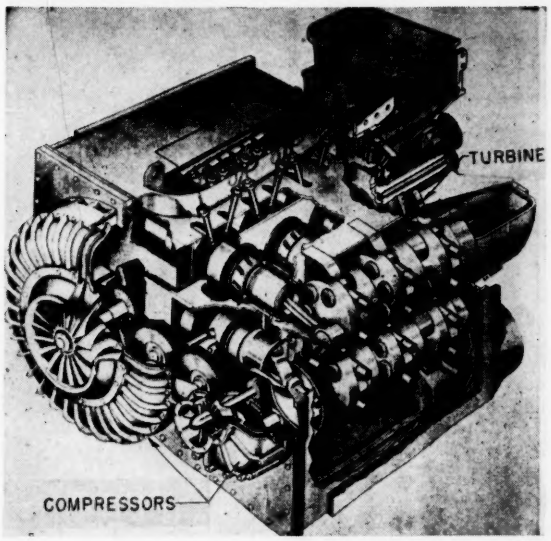
The one example of this class is the Orion engine, Fig. 5(*a*), developed by the General Electric Company. This development was sponsored by the U. S. Army Ordnance Corps for the tank engine in the June, 1950, to December, 1955, period. It is an air-cooled 2-bank, 6-cyl opposed-piston uniflow-scavenged engine, with four 3-throw crankshafts, two centrifugal compressors, and the associated gearing coupling them to the crankshafts. A free-power turbine provides the net shaft output of the system. One of the centrifugal compressors, the larger one in Fig. 5(*b*), provides the cooling-air flow at 2.1 atm pressure; the other compressor provides the engine-air flow at roughly 2.3 atm pressure. The exhaust from the cylinders combines with the cooling-air flow at a pressure of about 1.8 atm and then exhausts through the power turbine. Because the cooling air is utilized in the power turbine, this engine operates virtually adiabatically in contrast to the usual 15 to 25 per cent heat-rejection figure for most piston internal-combustion engines. The supercharge pressure ratio of 2.1 to 2.3 is, however, low relative to the Napier Deltic and Nomad engines and also low relative to the free-piston-type engines to be described.

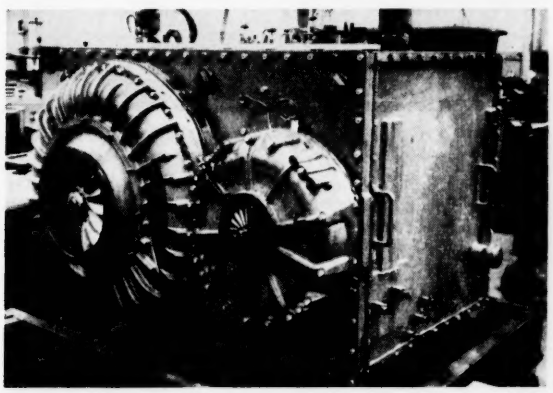
The achievement of adequate air-cooling of a highly supercharged diesel cylinder is a particularly noteworthy accomplishment. It represents refined techniques which could be applied to good advantage in most high-rating cylinder designs. The novel development of an accumulator-type injection system successfully solved the difficult aspects of high-speed high-rate timing and atomization.

Unfortunately, cancellation of this program at the first prototype stage has prevented a definitive evaluation of this engine concept.

Free-Piston-and-Turbine Engine With Turbo-Power Output

Thermodynamically, this class of engine, Fig. 1, is identical to the previously considered crank-piston-and-turbine engine with turbo-power output. All the piston-engine work is absorbed by the compressor and the power turbine is connected only to the load. Mechanically, however, there is a great difference. The compressor is a piston type rather than a centrifugal or axial-flow type. In consequence, the compressor piston can be directly driven by the engine piston without gearing and even without a crankshaft. Therefore remarkable mechanical sim-





Figs. 5 (*a* and *b*) The Orion Rigel engine, compression-ignition, air-cooled, 6-cyl opposed pistons; 4 crankshafts; two centrifugal compressors and one-stage axial-flow power-turbine; 600 shp

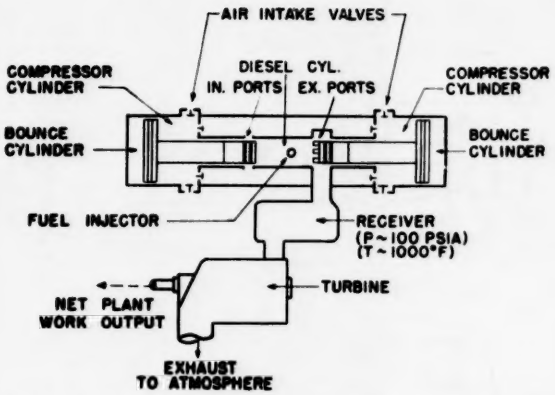


Fig. 6 Configuration of a free-piston-and-turbine compound engine

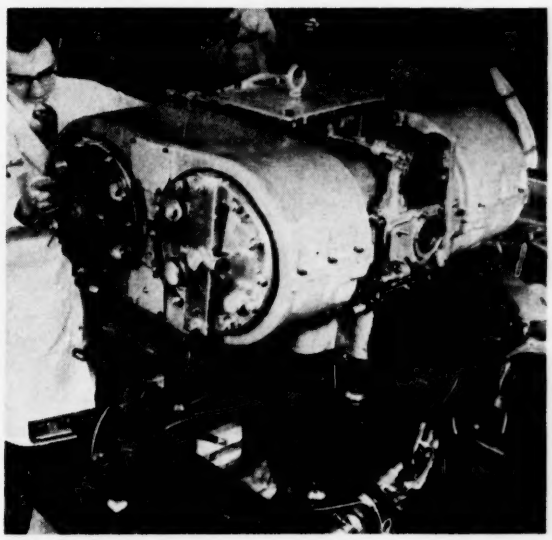
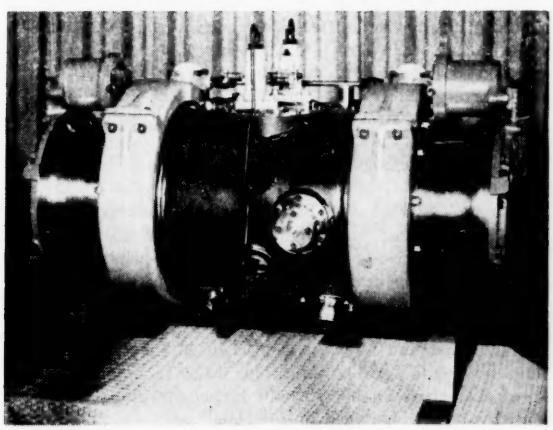
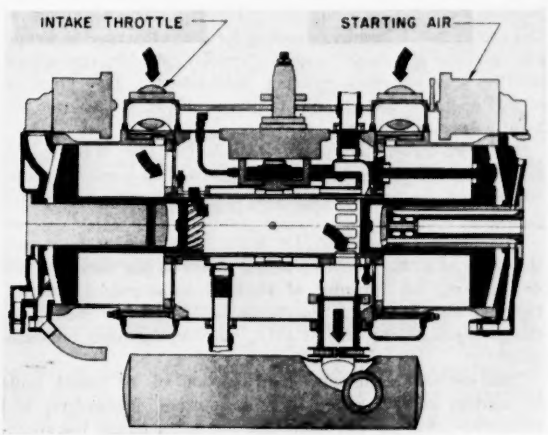


Fig. 7 The GM Hyprex GMR 4-4 engine: Two-cyl, 4-piston, Siamese-twin configuration; rated 250 shp at 4 : 1 supercharge (potentially 500 hp at 6:1)





Figs. 8 (*a* and *b*) Ford Model 519 free-piston gas generator with intake throttling for light-load operation

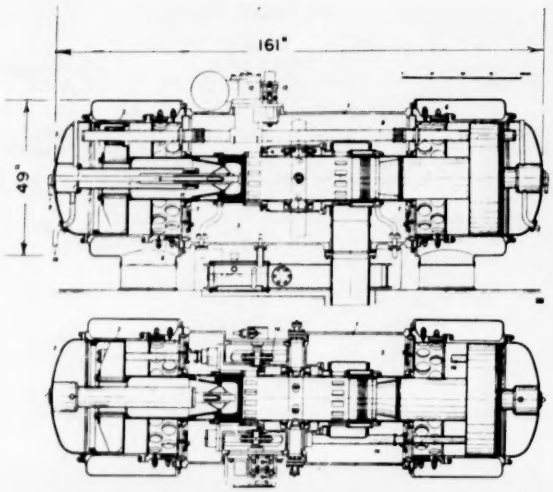


Fig. 9 SIGMA Model GS-34 unit with 1000-shp rating

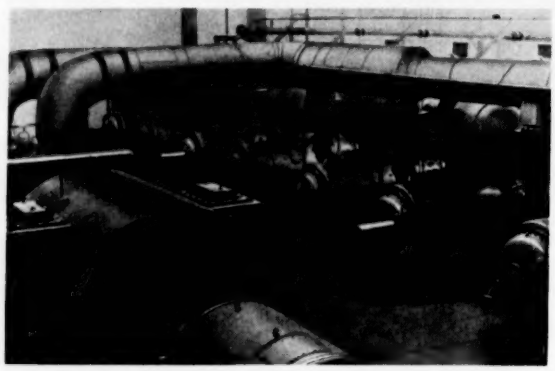


Fig. 10 Cherbourg Central Station Model GS-34 Installation: Eight gas generators driving a gas turbine; 6000-kw output

plicity results from this engine as can be seen in Fig. 6 where an opposed-piston configuration is described.

Ford and General Motors have automotive-type units under development. A unique aspect of the GM Hyprex GMR 4-4 engine, Fig. 7, is the Siamese-twin coupling of two cylinders to provide a common scavenge-space housing. The advantage of this arrangement, besides providing for the better specific-weight characteristic of a larger number of smaller cylinders, comes from smaller scavenge-space pressure fluctuations. The potential of this engine is 250 hp at a pressure ratio of 4, and over 500 hp at a pressure ratio of 6.

The *Ford* automotive unit is described in Figs. 8 (*a* and *b*). Unique aspects of this engine are the provisions made for piston cooling and the use of compressor-intake throttling to achieved light-load operation.

The 1000-hp *French SIGMA Model GS-34*, Fig. 9, represents the fruits of a 25-yr developmental effort on the part of Rault de Pascara, the inventor of the free-piston-engine concept; together with R. Huber, the Technical Director of SEME, the design organization; and SIGMA, the manufacturing organization.

The successful commercial application of 90 GS-34 units in marine, central-station, pumping-station, locomotive, and industrial-power service, more than any other factor, has stimulated the free-piston development in this country. A recent

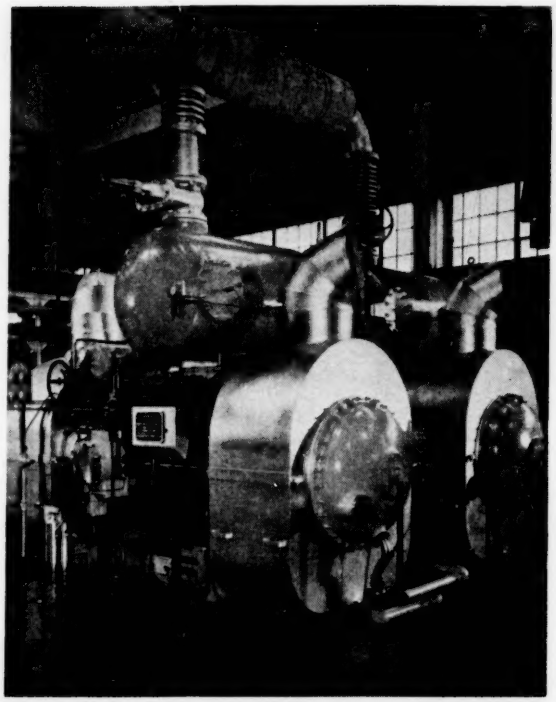


Fig. 11 GM (Cleveland Diesel Division) Model GM-14; six of these units installed on the converted Liberty ship *William Patterson*, 6000 shp

central-station application is described in Fig. 10. Eight gas generators feed a single gas turbine for a 6000-kw plant. General Motors has a co-operative arrangement with SIGMA and the Hyprex unit described previously received its preliminary design under Huber's direction. Another product of this arrangement is the model GM-14, Fig. 11. It is a direct outgrowth of the SIGMA GS-34 design with many improvements conceived by GM as a result of an extensive test program. Six of these were built by the Cleveland Diesel Division and are installed on the *William Patterson*, a converted Liberty ship. This ship had its trial runs in September, 1957.

These engines of the SIGMA GS-34 type operate successfully on residual fuel oil but must be started and stopped with a short run on distillate fuel.

The *Junkers* free-piston-diesel 4-stage air compressor, Fig. 12, served to introduce the free-piston concept in this country in 1943. There is no power turbine associated with it and the diesel engine operates with virtually no supercharge, just 4-in-Hg-gage scavenge-air boost is provided to clear and charge the cylinder for the 2-stroke cycle. This particular model was standard equipment in the German Navy (starting 1935) for providing 3000-psi air for charging the tanks used for torpedo launching. Junkers today markets a line of industrial free-piston compressors in Europe. SIGMA also has an industrial compressor on the market and, in fact, the commercial success of this compressor provided the "bread-and-butter" money which helped to support the long and trying development of the 1000-shp Model GS-34.

The following "advantages in principle" should be noted for the free-piston-type compound engine relative to the others, Fig. 1. First and foremost is mechanical simplicity; second, fluid coupling to a free-power turbine makes for excellent torque-speed characteristics and virtually immediate throttle response. The two "disadvantages in principle" are: (*a*) The engine cylinder should operate at a supercharge pressure greater than

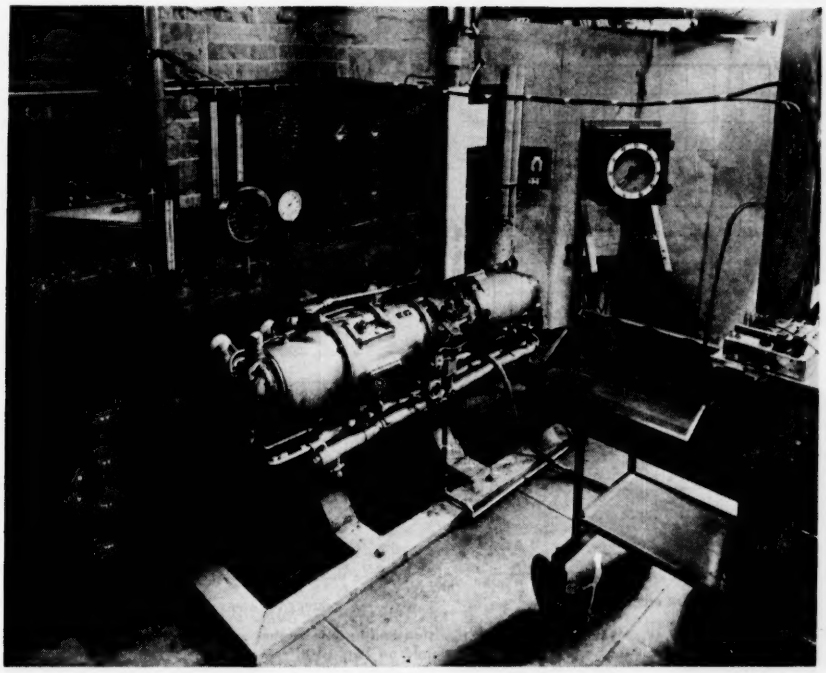


Fig. 12 German Junkers free-piston-diesel 4-stage air compressor

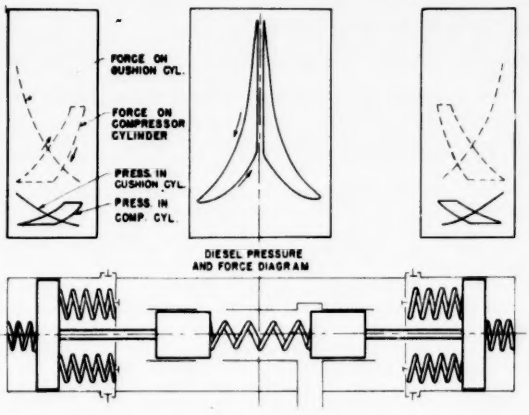

FORCE ON CUSHION CYL.
FORCE ON COMPRESSOR CYLINDER
PRESS. IN CUSHION CYL.
PRESS. IN COMP. CYL.
DIESEL PRESSURE AND FORCE DIAGRAM

Fig. 13 Spring-mass nature of the free-piston gas generator

3.5 atm in order to have a fuel economy competitive with a conventional diesel which in turn, brings the difficult "heat problems" associated with all highly supercharged piston engines; (*b*) light-load matching of the flow characteristics of piston and turbine engines is not readily achieved without resorting to wasteful throttling or blowoff methods.

The gas generator, in effect, is a spring-mass system which operates at a natural frequency established by the mass of the piston assemblies and the character of the gas springs, Fig. 13. There is absolutely no "flywheel" or rotating-inertia effect associated with this engine. Its response to a fuel-throttle change occurs within the period of one cycle. This close tie between the thermodynamics, establishing the character of the gas springs; and the dynamics, establishing the speed of operation; is one of the complicating design features of the free-piston system and, no doubt, has contributed to the delay on the part of crank-engine designers in accepting this engine concept.

Dynamical considerations become necessary only when the designer wishes to size a unit for a specified output. Thermodynamic analyses can be made of the cycle using well established procedures; moreover, these results will also apply to the crank-piston-and-turbine engine with turbo-power output as represented by the Orion engine concept.

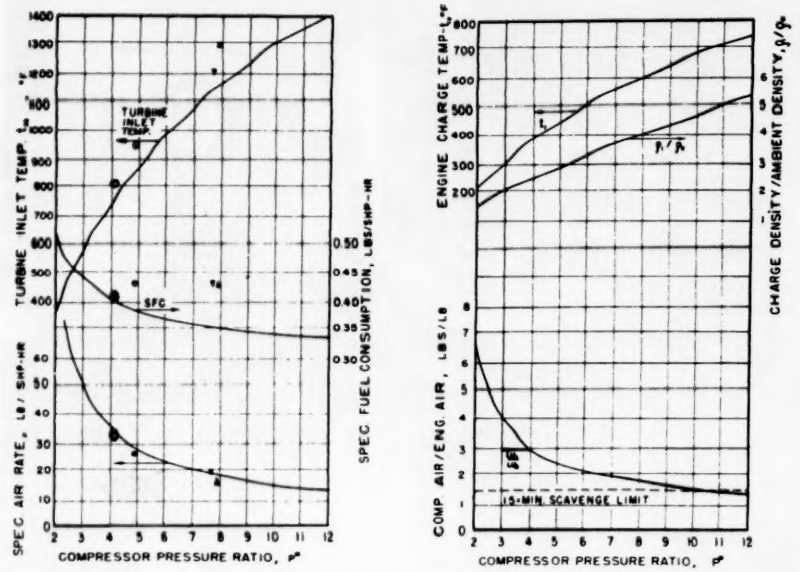
Figs. 14(*a* and *b*) provide a summary of some of these thermodynamic studies. Particularly noteworthy points are as follows:

(*a*) Modest turbine temperatures for supercharge pressure ratios less than 6, Fig. 14(*a*); (*b*) excellent specific fuel consumption for pressure ratios greater than 4, but poorer performance as pressure ratios approach 2, Fig. 14(*a*); (*c*) specific air rate intermediate to the diesel and simple-cycle gas-turbine engines, Fig. 14(*a*); (*d*) the high scavenge and combustion-air availability for pressure ratios less than 8, which together with high charge density permit operation with residual fuels as well as the conventional distillate fuel, Fig. 14(*b*); (*e*) the very high charge temperatures for pressure ratios greater than 4, Fig. 14(*a*).

This last factor has been the major contributor to the difficulty of the free-piston-engine development. Piston cylinder-heat problems, as characterized by ring failures and excessive wear, correlate directly with the charge inlet temperature. It is noteworthy in this respect that the early free-piston-engine designs aimed at high efficiency, by operation at high-pressure ratio, at the risk of insurmountable ring problems. In contrast, while SIGMA started with a pressure ratio of 6, they now rate their engine at a pressure ratio of about 4.5.

Summary and Conclusions

A remarkably substantial and diversified effort has gone into the crossbreeding of the piston engine with the turbine engine, Fig. 1. The prime objectives are to preserve the best features of each—the high thermal efficiency of the piston engine and the power compactness and torque-speed characteristics of the turbine engine. This has been achieved with both technical and commercial success in three of the four hybrids



Figs. 14(a and b) Predicted performance for the piston-and-turbine engine with turbo-power output applicable to both free-piston and crank types; data points on Fig. 14(a) represent actual test performance of early prototype free-piston units

shown, Fig. 1. In the case of the free-piston type even mechanical simplicity has been achieved.

Possibly the future will result in the creation of more members of the family. With considerable certainty, we can anticipate accelerated evolutionary progress now that the revolutionary changes have been accomplished, especially in view of increasing fuel costs and the pressing need to reduce or hold the line on capital costs by attaining improved power compactness and reduced specific weight.

Acknowledgments

In addition to leaning heavily on the papers presented in the accompanying selected bibliography, the author is indebted to R. Huber, E. Chatterton, R. J. Hooker, P. Klotsch, and Eric R. Brater for providing essential assistance in the way of illustrations and viewpoints.

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AVIATION

Introduction

THE nature of the development in aviation gas turbines during recent years has been essentially different from that presented in our first report in 1952. In this first period, the engines were new and revolutionary. A new type engine was developed which was destined to completely change the thinking in the aviation industry. This light power plant permitted high speed flights previously unattainable.

The first period of 13 years might be likened to the birth and early childhood. Like the early history of a new species, many of the engines died in their infancy and only a few survived the test of usefulness. Yet those which did survive brought a new light to the horizon. In fact, they were bright stars in the aviation heaven which served to guide the path to new conquests. Two new types of engines were born, the centrifugal and the axial-flow gas turbines, and both were adapted to deliver their power through a shaft (turboprop) or a high velocity jet (turbojet). In this period of infancy these engines were carefully nurtured. Every cry was answered, for the program received strong financial support from the various national governments.

In the past 6-year period, the one under discussion today, the engines were in their youth or "teen age" years. They were still growing in size but for the most part the development took on the usual teen age general refinement and sophistication. As is the case with our children, in this period, some were severed from the "motherly" financial support of the national government and were left to hustle on their own in the struggle for existence. Or stated in other terms some few of the engines were developed completely, or the development continued on private-company funds with the hope that these engines would be competitive.

The magnitude of the results of these refinements can be appreciated from a few specific examples. The Rolls Royce Dart, a turboprop, clearly illustrates the point. When it first ran in 1948 it had an equivalent shaft horsepower, eshp, of 1125 and a specific fuel consumption of 0.830 lb per hr per eshp. Since then, it has been improved step by step through revised models, with eshp of 1540, 1720, 1990, 2105, and 2655, and with corresponding specific fuel consumption of 0.753, 0.698, 0.680, 0.665, and 0.640, respectively. These improvements resulted from increased air flow and higher compression ratios without sacrificing but rather improving component efficiencies. There were, however, some sacrifices. The weight increased by about 250 lb although specific weight without a propeller was reduced from 0.890 to 0.473 lb per eshp.

The improvements made in the small-gas-turbine class, where the development of efficient engines is most difficult, are illustrated by the Boeing T-50. In 1948, this engine had a specific fuel consumption of 1.6, a compression ratio of 2.7, a compressor efficiency of 0.69, and a turbine efficiency of 0.67; by 1951, these values had been changed to 1.3, 3.0, and 0.72 and 0.80, respectively; by 1952, they were 1.1, 4.0, 0.72, and 0.83; and by 1957, they were 0.75, 6.6, 0.78, and 0.855. During this period the weight increased from 220 to 320 lb and the eshp went from 120 to 240, or exactly doubled, but the specific weight was reduced from 1.8 to 1.33.

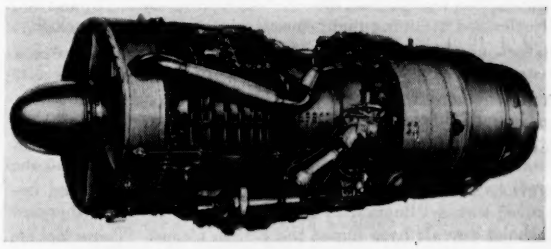

Fig. 1 Avon (Courtesy of Rolls-Royce Limited)

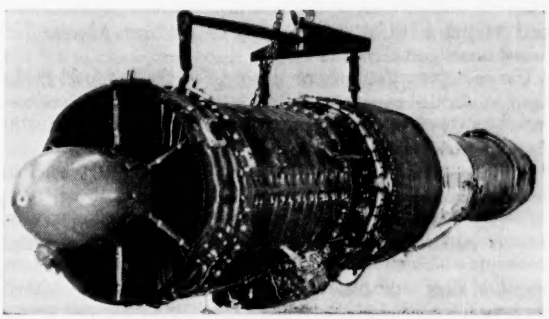

Fig. 2 ATAR—8 (Courtesy of Société Nationale d'Etude et de Construction de Moteurs d'Aviation)

In the turbojet area the examples of radical improvement are not as common during this period, for they have led the turboprops in development and many refinements were made during the first period as illustrated by the J-33, J-35, J-47, and Nene. Yet the newer engine which had had the advantages of this earlier experience also responded to engineering refinements as is illustrated by the Avon, Fig. 1. At initiation in 1947, it produced 6500 lb of static thrust with a specific fuel consumption of 0.9. Successive versions developed 7500, 8000, 9500, and 10,000 lb of thrust and at the same time the specific fuel consumption of the latter version was decreased to 0.84 lb per hr per lb of thrust.

Another example of the improvements which have been made is vividly expressed by Constant in his article on "Ten Years of ATAR Development." This French engine, Fig. 2, developed by the Société Nationale d'Etude et de Construction de Moteurs d'Aviation now has the lightest weight of any nonexpendable engine in its size. Constant shows the little modifications on the engine which produced the improvements. During the 8-yr period from 1948, the date of first tests, to 1956, the thrust was in-

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creased from 3750 to 9680 lb and at the same time the specific weight was reduced from 0.48 to 0.25, and the specific fuel consumption was reduced by 20 per cent.

The Gas Turbine's Place in Aviation

Even in the first period, gas-turbine engines were sufficiently developed to be given military application, and now all new military combat planes (fighter, attack, and bomber aircraft) employ these engines in some form or other, mainly turbojets. Instead of the high subsonic speeds produced by the early jet fighters, the new aircraft like the Starfighter, the P.1B, and the Hustler are attaining flight speeds higher than sonic velocity. In fact, one has a maximum flight velocity of Mach 2.5. A few years ago this would have seemed absurd, but now Mach 5 is considered an intermediate speed by some research workers.

Not only do gas turbines dominate the military field, but they are having an impact on commercial aviation. The British had established the first jet-transport service at the time of the previous report. After a few difficult years, the new and improved turbojet liners were a success, and now several other new turbojet aircraft have joined the British Comet. These include the large planes like the Boeing 707, the Douglas DC-8, and the Russian TU-104; the intermediate-size planes like the French Caravelle, and the Convair 880; and the small ones like the Lockheed Jet Star. The latter carries only 10 passengers whereas the largest mentioned carries 144.

The turboprop planes have also entered the transport field. Again the British were first. On April 19, 1953, the British European Airways established the first regular commercial airlines with planes powered by turboprops, the Viscounts propelled by Rolls Royce Darts. The immediate success of these planes on the route from London to Cypress caused expansion to other routes. On June 26, 1955, the Capitol Airlines established the first regular commercial turboprop air route in the United States employing Viscounts with improved Dart engines. Within a year they were operating 20 of these planes on eight different airlines. Another version of the same plane was also put into commercial use in Canada. To date over 327 Viscounts have been sold.

Following the Viscounts in the commercial turboprop field were the Bristol Britannia powered by the Proteus, the Lockheed Electra propelled by the Allison T-56, Fig. 3, and the Russian TU-114 propelled by NK-12N turboprop. All three are designed for longer routes than the Viscounts. The Vanguard, a replacement for the Viscounts, and the Convair Metropolitan will soon come into the picture.

Most of the airlines agree that the gas-turbine is the power plant of the future. Pure jets will probably be used chiefly for first-class long-distance services and turboprops for short and medium distances, for tourist rates, and for third-class service. The list of ordered airplanes of the principal airlines of the world given in *Interavia*, September, 1956, shows that of the 936 airplanes on order, 344 were propelled by turboprops and 279 by turbojets. Thus about two thirds of the planes will employ gas turbines, over half of which are turboprops.

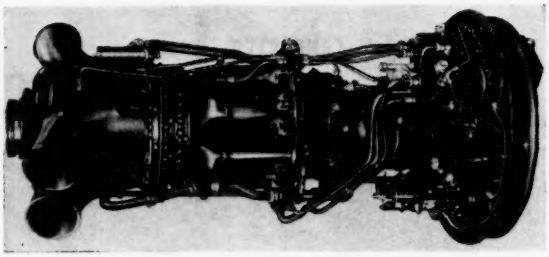


Fig. 4 Gazelle (Courtesy of D. Napier & Son Limited)

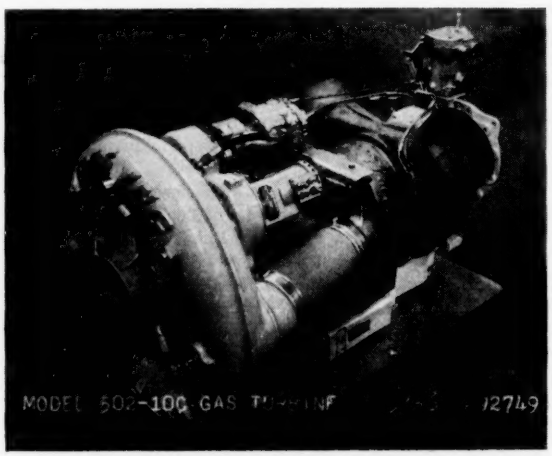


Fig. 5 T-50-1 (Courtesy of Boeing Aircraft Company)

During this period the gas turbines broke into a new field—power plants for helicopters. Driggs and Lancaster stated in 1952 that the "free turbine" turboprop is the "only" logical engine for helicopters. This was based on the idea that for optimum helicopter use several gear ratios are required, and the degree of freedom of a free turbine is equivalent to a continuum of gear ratios. Evidently many other people agreed since several engines are now being developed especially for helicopter operation; namely the T-58, the T-53, the Gazelle, Fig. 4, one version of the Eland, the Palouste IV, the Artouste II, and the Turmo III. Engines have been tested in the Sikorsky S-58, the Vertol H-21D, the Wessex, and the Bell XH-40.

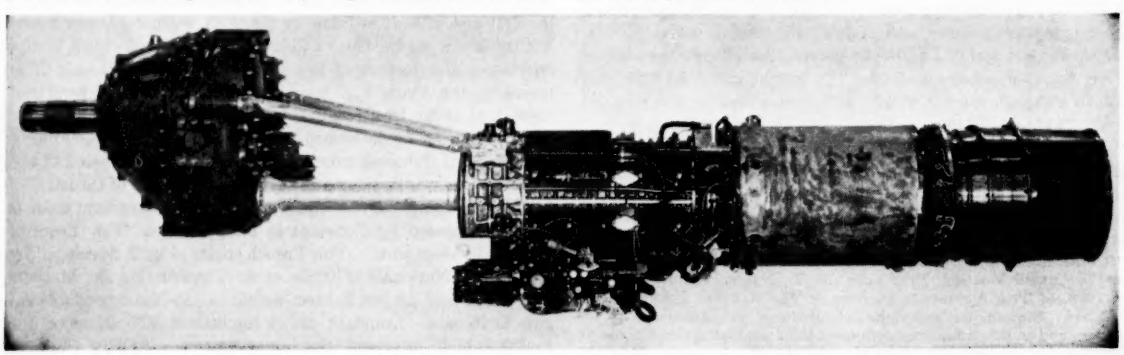


Fig. 3 T-56-A (Courtesy of Allison Division of General Motors)

Another new area of encroachment of the gas-turbine engines on the aviation industry was in the field of light airplanes. A Cessna L-19B is now propelled by the Boeing T-50 (Fig. 5). The next progress report will undoubtedly contain information about important advances in this area.

Effects of Gas Turbines on Aircraft Design

The influence of gas-turbine power plants has not been limited to increased flight speed. Military aircraft will be operational up to 60,000 ft and perhaps more. Commercial transport planes will cruise at altitudes from 30,000 to 45,000 ft, depending upon the distance and fuel load. These greater possible flight altitudes have resulted from the fact that the additional lift, produced by the increased flight speed, more than offset the reduction in lift due to the lower air density. It is logical to fly at these higher altitudes because of the decrease in fuel consumption, which at 40,000 ft is approximately one fifth that at sea level. As a result of the reduction in fuel consumption with altitude, almost all planes will have a climbing cruise. This will develop problems of navigation and control of flights.

The increases in altitude and speed created many new research problems for the aircraft designer. High aircraft speeds were attainable by the use of the light powerful jet engines without any essential changes in the air frames. The airplane industry was not willing to stop there. Numerous aerodynamic studies were begun to determine the characteristics of many wing sections, body shapes, aircraft planforms, and configurations at the higher velocities. In order to reduce the drag, wings were made as thin as possible and with sharp leading edges. Swept wings and delta wings were exploited. The aspect ratios have decreased from 8 or 6 to as low as 1. The wing loadings have increased from about 40 psf to over 100 psf. If the discussed wingless aircraft materialize, our classical methods of performance analysis based upon wing loading must be modified.

In order to help determine the combination of fuselage and wing with minimum drag, an area-rule concept was developed. This led to planes with concave areas in the fuselage in the neighborhood of the juncture of the wings.

Stability and control problems had to be investigated thoroughly. Aerodynamic moments near sonic velocity, and in the rarefied air were of special interest.

Higher flight speed made possible higher wing loading (lift is proportional to velocity squared), or aircraft with less wing area. Yet at take-off the same lift was required for a given weight. In order to get this lift, either the take-off (and landing) speed, or the life coefficient, or both, had to be increased. Even though turbojet engines are very powerful at high velocities, their static thrust is much less than that for reciprocating engines. (Engines of the same weight would be comparable but the jet engines employed must be much lighter to be competitive). Thus the acceleration of the aircraft along the runways is low. Both of these effects, high wing loading and low acceleration, add up to longer and longer runways, although a variety of methods for reducing the runways have been proposed. Attinello conceived the fruitful idea called supercirculation using compressor bleed air to create supersonic jets blowing over the airplane flaps, thereby increasing the circulation about the wing. This system not only delays separation but increases the lift coefficient at all angles of attack and has now been installed on the latest fighter aircraft in the U. S. and England. Driggs, Hyatt, and Ellis expounded the idea of a vertical take-off and landing aircraft. Such planes are still being developed and are known as VTOL planes. Both turboprops and turbojets have been used in these designs. Designs for short take-off or landing have also been suggested. (Length of runways was never a problem for turboprop airplanes for these engines give more static thrust than corresponding reciprocating engines.)

High flight velocities have introduced the hazards of aerodynamic heating, and ways of absorbing, transferring, or avoiding this heat are still being studied.

The topics mentioned and other interesting experiments have led to the development of costly supersonic wind tunnels and flight-testing equipment.

The rapid rate of fuel consumption required at low altitudes has caused a change in flight plans. In the commercial flights there had to be a speeding up in take-off and landing procedures. Fuel could not be spared for long warm-ups nor were these necessary. Time could not be allowed for loiter before landing or to encircle storms. Extended weather information is needed both as to range and altitudes and reduced time for navigation permits fewer fixes.

Noise is a major problem, however, and so much research has been done on noise suppressors that it has been predicted that by the time commercial jet service is established the noise output will be no worse than the output for piston-engine airplanes.

Effects of Aircraft on Gas-Turbine Engines

These changes in aircraft trends resulting from the use of gas turbines have in turn had their effect on the gas turbines themselves.

The high velocities have required more and more thrust which has led to larger and larger engines and increased the requirements for afterburning and other means of augmentation. The highest static thrust listed in the previous report for a single jet-engine unit was less than 10,000 lb and there was only one near this magnitude. Now there are units with static thrust of 20,000 lb without augmentation, and over 25,000 lb with afterburning.

At first there was a tendency to increase the compression ratios of the rotating compressors in order to increase the cycle efficiency and hence, reduce the specific fuel consumption. Later, on the engines designed for the highest velocities, the design compression ratios of the rotating machinery were reduced because the high ram pressures coupled with small mechanical compression gave more optimum performance for current operating temperatures.

These higher velocities, however, put more stringent requirements on the inlet and diffuser in order to develop efficient recovery of the ram pressure. Because of the several speeds of flight, variable-area diffuser inlets are desirable.

The use of bleed air for lift required that engines be equipped with bleed outlet and, what is more important, designed for efficient operation with and without bleed.

Methods for using the engine power to reduce landing distance have been explored.

The use of the engines in the vertical position for longer than a few seconds required the development of a lubricating system, which would supply the bearings in any engine orientation.

The higher and higher flight altitudes placed added restrictions on the burners to insure efficient combustion in the rarefied air. Also at these altitudes Reynolds-number effects became more important in the designs.

Thus we see that in this period the impact that jet engines had on aircraft in turn created an impact on the jet-engine development. It is expected that this will continue and that the two will develop hand in hand for many years to come, as was the case with airplanes and reciprocating engines in the first 40 years of heavier-than-air flight.

Major Engine Changes

Because of the greater simplicity and lighter weight, in the first period, the centrifugal-compressor engines were the first to propel aircraft and they were produced in the greatest numbers. The centrifugal-compression type soon began to fall by the way-

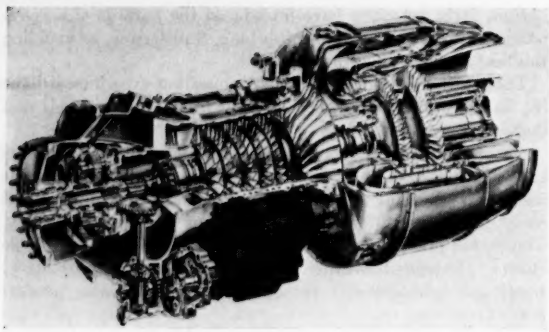


Fig. 6 T-53 (Courtesy of Lycoming Division of Avco Manufacturing Corporation)

side. Except in the small-engine field, practically no new engines of this type were developed. It was recognized from the start that axial-flow engines definitely had greater potential from the standpoint of specific fuel consumption and frontal area. So it was only natural that during this period of refinement and sophistication axial-compressor engines should receive the impetus.

The French Verdon, the only large centrifugal-compressor-type turbojet introduced during this period, appeared in the first year. The Verdon is a large engine in its class (intermediate in size when axial-compressor engines are included) with a static sea-level air flow of 154 lb per sec and a dry thrust of 7700 lb. Although this engine had a large air flow and thrust per unit of frontal area, the *significant characteristic* was its low specific weight of 0.267 lb per lb of thrust. This is about 17½ per cent less than that of any engine mentioned in the first report, and the Verdon is a regular engine, not the expendable type. Unfortunately the specific fuel consumption was as high, if not higher, than on the preceding engines of this type.

The centrifugal compressors were not completely discarded. Most of the smaller engines incorporated a plan, started in the first period by the Bristol Proteus, of combining both axial and centrifugal units, by placing a single centrifugal stage after a series of axial stages. The small turbojets, the Gourden, the Gabizo and the Soular; and the turboprops, the T-53 (Fig. 6), the T-55, the Artouste, the Astazou, the Bastan, and the Turmo were all of this type. In this rear location the centrifugal compressor can give a large increment in pressure without essentially affecting the frontal area, because of the reduced air volume resulting from the precompression of the axial stages. Moreover, it helps to minimize the Reynolds-number effect which becomes critical on small axial compressors at high altitudes.

There were two paramount changes in the axial-flow engine designs during this period: (*a*) The development of two-spool engines, and (*b*) the perfection of movable-stator compressor blades. Both resulted from efforts to increase the compression ratios to fully exploit the potential of the axial-flow engine for lower specific fuel consumption. These higher compression ratios, in turn, amplified the difficulties by increasing the probability of encountering the surge line on starting and accelerating from idle to normal rotational speed, and of obtaining lower efficiencies at partial powers.

The development of the first of these—the two-spool engine—was actually begun in the first period but will be discussed here since performance information was not available for the first report. In a two-spool engine some of the rear banks of the rotating compressor blades and some of the front banks of the rotating turbine blades are connected to one shaft. The remainder of the rotating blades of the compressor and turbine are connected to a concentric shaft passing inside the first one and rotating independently of it. The improved flexibility of operation makes it one of the most attractive designs. The two-spool engine not only relieves the starting-acceleration problems but permits a high pressure rise per stage of compressor. This improved performance per stage is not, however, enough to offset the weight penalty for the two shafts. The J-57, Fig. 7, the Olympus, the J-67, the Iroquois, and the Conway are all of the two-spool types.

In an engine with variable-compressor stator blades, as the name implies, the angles of the nonrotating compressor blades may be changed to keep the angle of attack of the compressor blades small enough to prevent stall, which is particularly important in the starting regions. For starting, the blades will be closed as much as 30 deg and then slowly opened as the rpm is increased until the design angles are reached. A schedule for the changing of the stator-blade angles may be programmed into the engine-control system. Most of the development on movable compressor blades has been done by the General Electric Company. The J-79, Fig. 8, and the T-58 both have incorporated this feature in their designs. It has been said that the variable blades in the J-79 render the flow breakaway in the compressor practically impossible, and that they insure optimum compression ratio throughout the speed range.

A third way to avoid the surge upon starting is to bleed air from the compressor. The J-54, Fig. 9, has an automatic-bleed program built into the controls.

Another milestone in the period was the construction of a by-

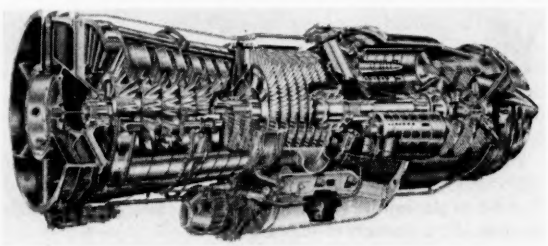


Fig. 7 J-57-P (Courtesy of Pratt & Whitney Aircraft)

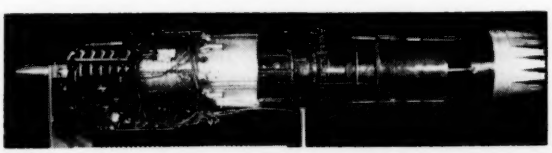


Fig. 8 J-79-GE (Courtesy of General Electric Company)

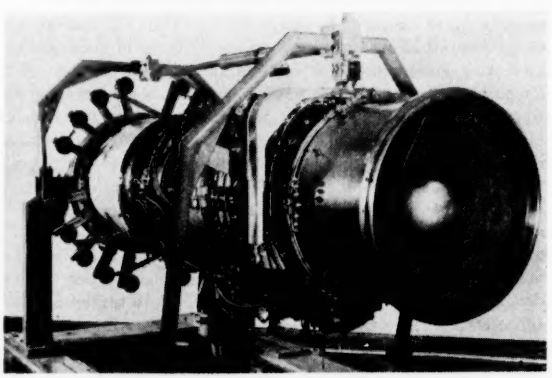


Fig. 9 J-54-WE (Courtesy of Westinghouse Electric Corporation)

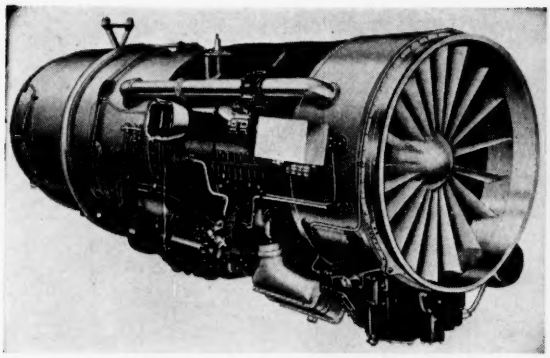


Fig. 10 Conway (Courtesy of Rolls-Royce Limited)

pass engine, Fig. 10. Immediately after the success of the first turbojets, the possible advantages of using some of the jet energy to activate a larger mass of air were recognized. In spite of the additional complications and weight increases resulting from the extra turbine stages, the gear box, and the propeller, the turboprops were developed to take advantage of the high propulsive efficiencies. The question still remains whether there is a place for an intermediate-type engine between the turbojet and the turbprop where a smaller mass of air than usual is associated with a propeller. During the first period, the British tried to fill this intermediate area by fans, which basically consisted of blades extending from the outer casing of the turbine, and attached to the turbine, and rotated with it. In the F-3, the fan was inclosed by an external shroud or casing and called a ducted fan. Except for the mixing of the secondary mass with the internal mass flowing throughout the engine, the engine functioned much like a fixed-pitch propeller. The F-3 and the unducted fan, F-5, are no longer in existence. During the past six years the British tried again to develop a powerful intermediate engine—the Rolls Royce Conway —a bypass engine. This differs from the ducted fan in that the compressor blades of the low-pressure spool are extended to compress and augment a secondary mass of air which does not pass through the high-pressure spool or the low-pressure turbine. This engine has one definite advantage over the ducted fan or a small propeller—very high thrusts can be obtained by employing burning in the auxiliary stream whose pressure is greater than ram pressure. Moreover, it is claimed that the mixing of the slower outer mass with the central high-velocity stream not only improves propulsive efficiency but reduces noise level. It has the advantage over the conventional turboprop at high speed in that the air may be diffused to lower the relative velocities.

Another first for this period was the introduction of a supercharged gas turbine. The Bristol Aero Engines Limited pioneered with their Orion, Figs. 11 and 12, which is designed to develop the same eshp from sea level to 20,000 ft. Above 20,000 ft, there is the usual drop-off in power. The engine is a normal two-spool gas generator with a coupled turbine; that is, the low-pressure turbine, in addition to driving the low-pressure compressor stages, drives the propeller. Obviously, such an engine could produce about 1.8 times as much power at sea level as it could at 20,000 ft. However, following supercharged-piston-engine practice, the engine controls are scheduled to limit the full-throttle power, thereby avoiding higher pressure stresses on the casings, thus permitting some weight reduction. In addition the weight of the reduction gear and propeller were materially less than would have been required for the maximum possible power. By this procedure the designers have achieved a high-power engine at the usual flight altitudes for a small bulk and weight, and a specific fuel consumption of 0.4 at 40,000 ft.

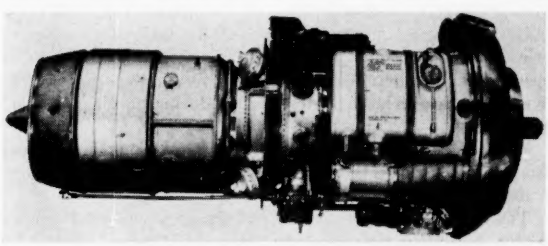


Fig. 11 Orion (Courtesy of Bristol Aero-Engines Limited)

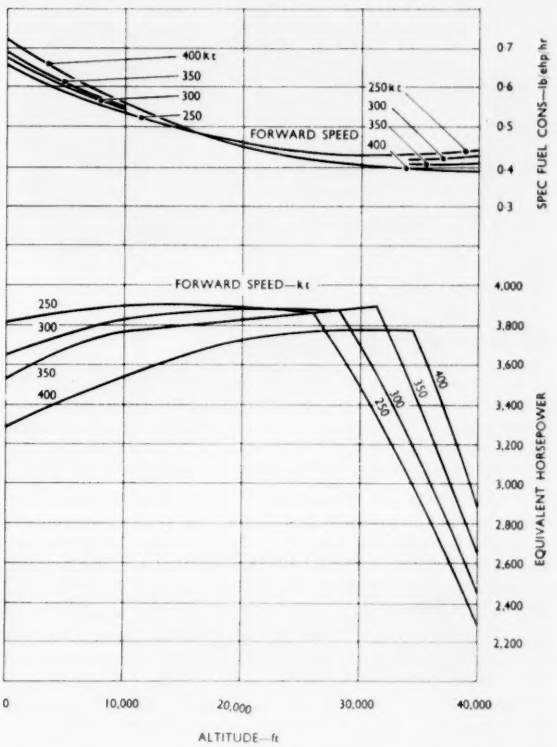


Fig. 12 Power and specific fuel consumption of the Orion (Courtesy of Bristol Aero-Engines Limited)

The use of gas turbines for helicopters has taken three directions: (*a*) Hot-gas generators, (*b*) gas compressor and pump, and (*c*) turboprops. In the hot-gas-generators system, of which the Napier Oryx is an example, the basic turbocompressor unit produces gases at an elevated pressure and temperature which are exhausted backward at the tips of the rotor. In the gas-compressor-pump system air is bled off after the compressor and is ducted to the rotor tips and mixed with fuel which is burned before the gases are expanded through nozzles. Here the effort has been mainly confined to analytical studies. When a turboprop is the main power plant the rotor is driven by a shaft through a gear train in the conventional piston-engine manner. Both the fixed and free turbines have been employed for this purpose, although the free turbine has the most apparent advantages and has received the most development attention. Many engines have been designed for this purpose including the Turmo, the Eland, the Gazelle, the T-58, the T-53, and the T-55. The Artouste is in the fixed-turbine class.

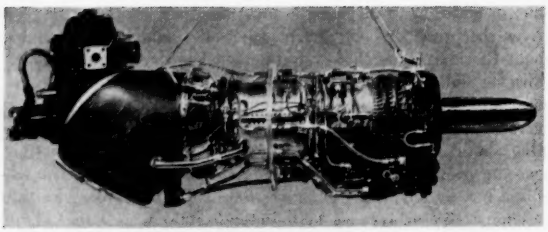


Fig. 13 T-58-GE (Courtesy of General Electric Company)

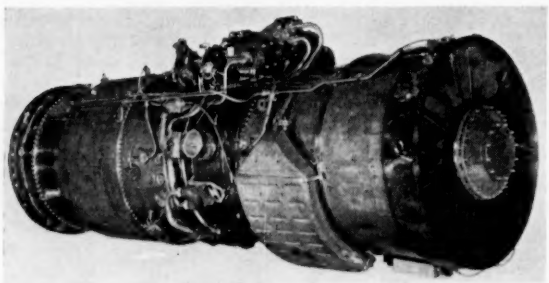


Fig. 14 J-75-P-2 (Courtesy of Pratt & Whitney Aircraft)

Activities of Principal Companies

Perhaps one of the best ways to get a picture of the general progress is to follow the developments of the various engine manufacturers. In the United States, for the most part, the same companies mentioned in the first report continued to design and test new engines. The General Electric Company divided their efforts into large and small-aircraft engine departments. The large-engine department at Evendale, Ohio, concentrates on engines with mass flows of over 60 lb of air per sec. They designed and built a J-53, a very large engine, which was canceled because of its heavy weight, although its compressor was later used to absorb power for propeller developments. It did, however, form the basis for the design of a scaled model, the J-79, a light, powerful, variable-stator jet engine with 12,000-lb static thrust, mentioned before. They designed and built the J-73, a smaller engine of 9200-lb thrust, which is no longer active. And no hardware was even built for their J-77 design. Currently they are concentrating on a new, larger engine. The small-engine department designed a light-weight expendable engine, the J-85, which produces a thrust of 2400 lb, the T-58, Fig. 13, a small variable-stator-blade turboprop for helicopters, with 1024 eshp; and the T-64, a larger helicopter turboprop with 2650 eshp.

The Pratt and Whitney Company continued to produce and improve the J-57, and made a smaller version of same design, the J-52, with a thrust of 7500 lb. Then the large J-75, Fig. 14, with a static thrust of 15,000 lb was built. In addition they have designed two other larger engines, one being for very high flight speed. In the turboprop area they toiled with a T-57, an adaptation of the J-57; and designed the T-52, which was canceled in favor of the J-52. They still have the old T-34, but their main progress has been limited to the turbojet field. The J-57, America's first two-spool engine, which did not receive strong government support because of some early disappointments, was perfected by Pratt and Whitney. After a long and careful test program coupled with carefully planned theoretical calculations an engine emerged which may be said to be the American jet engine of this era. It became a happy replacement for engines in its class size like the J-40 which never quite came up to expectations. There were more sales of this engine during the period than any other and perhaps as many as for all other jets combined. In this sense it played the role in this period occupied by the J-33 and J-47 in the first period. It powers the Boeing B-52, the North American F-100, the Convair F-102, the Boeing 707, the McDonald F-101, the Douglas F4D-1, the Douglas A3D, and the Chance-Vought F80-1.

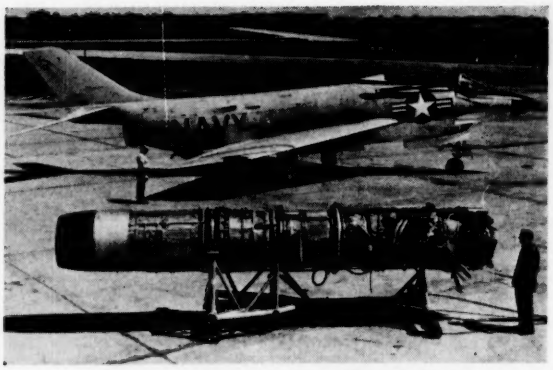


Fig. 15 J-71-A-2 (Courtesy of Allison Division of General Motors)

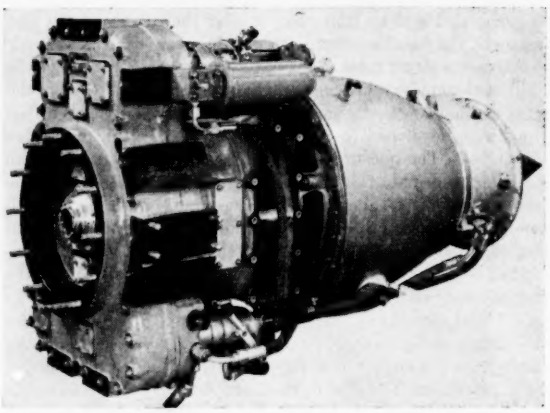


Fig. 16 T-51-T-3 (Courtesy of Continental Aviation & Engineering Corporation)

The Allison Division of General Motors continued development work on the T-56, a 3750-eshp turboprop, and did some work on a coupled version of this unit designated as the T-54. They have also designed a 5500-eshp two-spool turboprop known as the 550 or T-61. They produced a turbojet in the 10,000-lb class, the J-71, Fig. 15, and made design studies for two 25,000-lb turbojets, that have now been abandoned.

The Westinghouse Corporation moved their aviation-gas-turbine division to Kansas City, Mo., where they tried to perfect their J-46 which was an improved version of the J-34 with static thrust of 3860 lb. They continued some work on the J-50, which was a high-powered version of the J-40. Then, largely on their own initiative, they designed and built the J-54, their only successful engine after the J-34. This engine has a thrust of 6700 lb, a specific fuel consumption of 0.85, and a very low specific weight. Westinghouse also has a contract for producing the English Soar, a small engine with a thrust of 1860 lb.

Except for the production of the J-65, the Wright Aeronautical Company's efforts in the aviation-gas-turbine field were not very fruitful.

The Frederick Fladder Company designed a turbojet, the J-55, with a supersonic compressor, which ran but never met the requirements.

Fig. 17 T-62 Mock-up (Courtesy of Solar Aircraft Company)

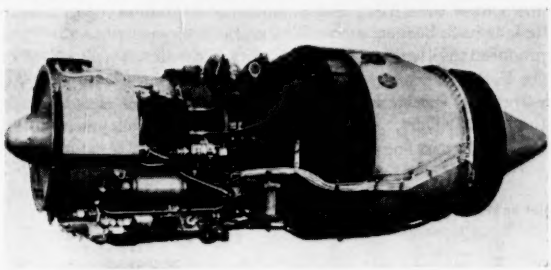


Fig. 18 Gabizo (Courtesy of Société Nationale d'Etude et de Construction de Moteurs d'Aviation)

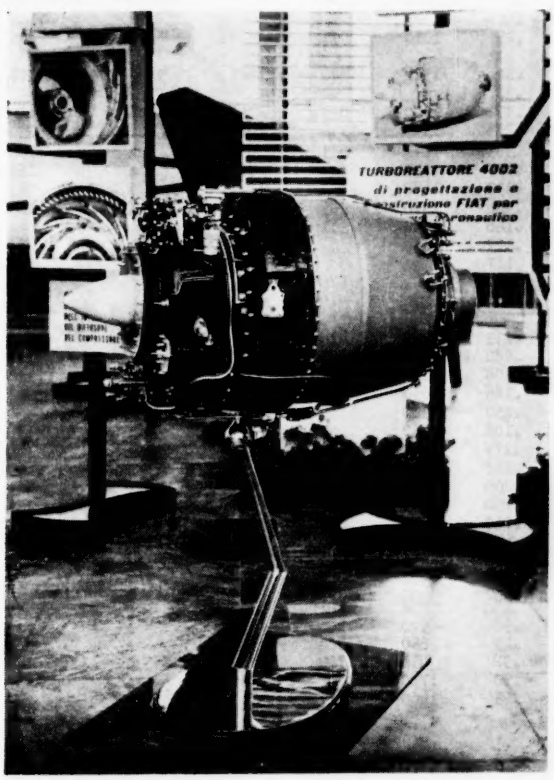


Fig. 19 Fiat (Courtesy of Fiat Divisione Aviazione)

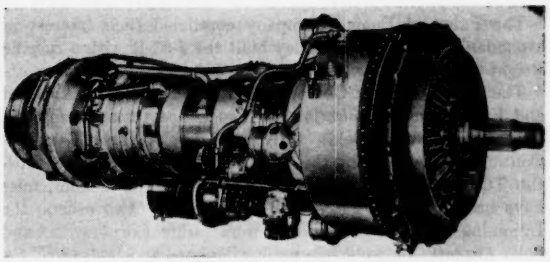


Fig. 20 Tyne (Courtesy of Rolls-Royce Limited)

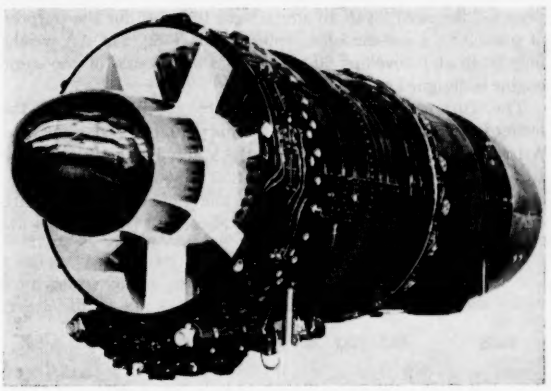


Fig. 21 Gyron (Courtesy of The DeHavilland Engine Company Limited)

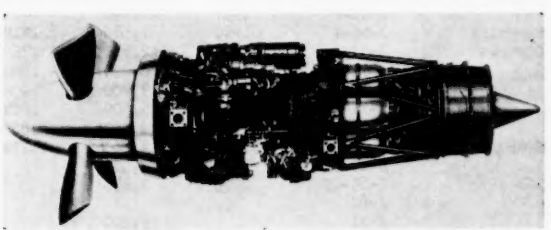


Fig. 22 Eland (Courtesy of D. Napier & Son Limited)

Fig. 23 Iroquois (Courtesy of the Orenda Engines Limited)

The Fairchild Engine Company continued their interest in expendable-type engines. They built the J-83-R with a specific weight of 0.122.

The Boeing Aircraft Company and The Continental Aviation and Engineering Corporation have sustained their activities in the small-gas-turbine field. Boeing has developed the T-60, and Continental is producing the T-51, Fig. 16, under contract with the Turboméca Company in France. These two companies have been joined in this small-engine area by two others, the Lycoming Division of Avco Manufacturing Corporation, and Solar Aircraft Company. Lycoming designed and built the T-53, the first turboprop specifically designed for helicopters. It has accumulated more flight time than any other engine of its type in the United States. They also designed another turboprop, for the same purpose, of about twice the size, the T-55. Solar Aircraft has designed the smallest of all aircraft gas turbines, for the purpose of propelling a one-man helicopter. This T-62, Fig. 17, weighs only 50 lb and develops 60 eshp: Another version of the same engine is designed as the T-66.

The Turboméca Company in France continued to be the leader in the small-engine field. In this period they designed the Artouste, the Astazou, the Bastan, the Marcudau, and the Turmo turboprops with eshp ranging from 350 to 800, and the Arbizon, the Gourden, the Gabizo, Fig. 18, and Soular turbojets with thrusts ranging from 550 to 2400 lb. Several of their engines are licensed for production by Blackburn and General Aircraft, Ltd., in England, and by Continental Aviation in the U. S.

The SNECMA in France remained mainly interested in the medium-size axial-flow-compressor engines and have done a remarkable job on the development of the ATAR, both from the standpoint of low weight and thrust per frontal area. The same could be said for the Hispano-Suiza Company on the Verdon. Both engines have been previously mentioned. The Hispano-Suiza has also assisted the Spanish in the design of their INI-11 turbojet. The Dassault Company is producing their R-7 and the British Viper.

The Fiat Divisione Aviazione entered the gas-turbine field. Formerly they had produced British-designed engines. In this period they have designed and developed their own Fiat 4002.001, which produced 716-lb thrust, Fig. 19. They continued to produce the English Goblin and Ghost and they made parts for the J-35, J-47, and J-65.

In the turbojet realm the Rolls Royce concentrated on the refinement of their Avon and designed the small Soar. Again their designs have been so successful that both these engines are being produced in other countries, the Avon in Sweden, and the Soar in the U. S. In the turboprop field they designed the Tyne, Fig. 20, with a low specific fuel consumption and a power of over twice that of their Dart. And they pioneered in the development of the Conway bypass engine.

Table 1 Turbojet engines

NAME	DESIGNATION	YEAR	F	$\frac{W_F}{F}$	$\frac{W}{F}$	$\frac{F}{A}$	$\frac{F}{L}$	C.R.	TYPE COMPRESSOR	W_a
	J-40-WE	1951	7,500	.95N	.4	1060	600			
	J-46-WE	1952	3,980		.467					
	J-54-WE	1956	6,700	.85	.24	1010	597	9	A + A	105
	J-55-WE	1948								
	J-57-P	1949	10,000	.80N	.42	1148	920	12.5	9A + 7A	170
	J-58-P	1957	18,500		.32					
Sapphire	J-65-W	1950	7,800	.91N	.351	1310	835	7	A	120
Olympus	J-67-W	1952	12,500	.75	.288	1390	1135	10.5	A + A	230
	J-71-A	1952	10,000	.80N	.383	1179	633	8	A	160
	J-73-GE	1952	9,200	.90N	.397	1250	744	7	A	150
	J-75-P	1954	15,000	.80N	.330	1470	850	12.5	A + A	253
	J-79-GE	1954	12,000	.80	.364	1378	555	12	A	165
Soar	J-81-WE	1955	1,860	1.27N	.148	1330	381			
	J-83-R	1956	2,400		.122			7	A	
	J-85-GE	1956	2,450		.117			7		
	J-87									
	J-91		25,000							
	J-93									
Viper	11	1951	2,460	1.11	.231	782	434	4	A	32
Sapphire	F-9	1948	11,000	.88	.280	1475	1221	7.3	A	176
Olympus	102	1950	12,000	.76N	.308	1250	1091	12	A + A	190
Orpheus	B.Or. 3	1954	4,850	.95N	.185	850	785	4.8	A	80
Gyron		1953	15,000	.85	.250	1300	1169	6	A	300
Gyron Jr.		1955	8,000		.188	1432		6	A	150
Avon	RA-26	1947	10,000	.86	.279	1040	1062	10	A	160
Soar		1954	1,810	1.27N	.148	1290	372		A	
Iroquois		1956	20,000	.85N	.200	1030	910	8.0	A + A	335
ATAR	8	1948	9,240	.90N	.250	1310	829	9	A	154
Verdon	R-450	1952	7,720	1.07	.267	586	895	4.9	C	130
Arbizon		1956	550	.97	.428	392	140	5.5		
Gourdon		1955	1,545	.96N	.250	551	357	5.0	A + C	
Gabizo		1955	2,400	.96	.244	632	465	5.2	A + C	
Soular		1956	705	.80	.502	400	143	5.0	A + C	
Fiat		1956	2,200	1.25N	.400	208	161	4.0	C	11
	M-012		6,600	1.06	.50	570	460	4.5	A	132
	M-018R		8,000	1.08	.49	650	488	7.0	A	160
	RD-45		6,170	1.09	.32	438	770	4.5	C	110
	M-205	1954	10,000	.9N	.30	1780			A	155
	M-209	1956	17,500	.85N	.25	1460			A	300

The Bristol-Aero Engines Limited designed and produced the Orpheus, a very lightweight turbojet with a thrust of 4850 lb, and a medium-size turbojet, the BE-47. Their novel endeavor was the supercharged turboprop, the Orion.

DeHavilland Engine Company, Ltd., concentrated on two simple single-spool supersonic engines the Gyron, Fig. 21, and the Gyron Junior, with thrusts of 20,000 and 7000, respectively, the first having a thrust at design speed of about 25,000 with afterburner.

During this period Napier and Son, Ltd., still concentrated on turboprops. They designed and perfected several models of the Eland, Fig. 22, increasing the eshp in stages from 3000 to 4200. They also designed and built the Oryx hot-gas generator, and the Gazelle turboprop for helicopters.

The Armstrong-Siddley Motors, Ltd., did not develop any new engines. The development on their Sapphire progressed along two lines, a high-temperature engine for high-speed military aircraft and a cool engine for transport applications. Their Viper continued in production in both England and France.

The Orenda Company in Canada has designed the largest of all the turbojets, the Iroquois with 20,000 lb of thrust, Fig. 23.

Although the author has had some correspondence concerning the developments in Russia, his information is too meager for him to make any concrete statements.

New Engine Summary

Tables 1 and 2 give some of the pertinent characteristics of the new turbojets and turboprops, respectively. The year denotes the date an engine was first tested. The performance characteristics are in most cases the most recent available data. Engines developed in the first period, whose performance was not available for publication in the first report are included and only the designations of some of the newest engines can be given.

F = static sea-level thrust, lb
F_c = thrust at cruise speed, lb
W_F = fuel flow static sea level, lb per hr
W_{Fc} = fuel at cruise speed, lb per hr
W = engine weight, lb
A = maximum frontal area, sq ft
L = length, ft
ESHP = static equivalent shaft horsepower sea level
$ESHP_c$ = cruise equivalent shaft horsepower
W_a = air flow, lb per sec
C.R. = compression ratio
A = axial compressor stages
C = centrifugal compressor stage
GHP = gas horsepower

The Armed Services of U. S. designate proposed turbojet engines by a J and turboprops by a T, followed by even numbers for engines developed under Navy sponsorship and odd numbers for those under Air Force sponsorship. In the first report even numbers for jets ran from 30 to 48, inclusive, and odd numbers from 31 to 73, inclusive. Now these numbers have been increased to 58 and 93, respectively, indicating five new jets under

Table 2 Turboprop engines

NAME	DESIGNATION	YEAR	ESHP	$\frac{W_F}{ESHP}$	$\frac{W}{ESHP}$	$\frac{ESHP}{A}$	$\frac{ESHP}{L}$	$\frac{W_{Fc}}{ESHP_c}$	C.R.	COMPRESSOR TYPE	W_a
Artouste	T-51-T-3	1954	425	.97	.555	192	141		3.9	C	7
	T-53-L	1954	850	.71N	.650	339	212		5.9	AC	10.5
	T-54-A	1954	7,500	.46N			622		9.25	A	72
	T-55-L	1955	1,650	.66	.417	524	333		6.3	AC	20
	T-56-A	1953	3,750	.46N	.47	430	311		9.25	A	36
	T-58-GE	1954	1,024	.64	.318	734	223		7.0	A	13.5
	T-60-S		400								
550	T-61-A		5,500	.51	.392			.385	10		
	T-61-S		60		.833						
	T-64-GE		2,650		.374				11	A	
	T-66-S		60								
Proteus	755	1951	4,120	.62	.693	486	496	.445	7.2	AC	42.5
Orion**	On 1	1955	5,150	.647	.613	541	551	.395	10	A + A	86
Eland	6	1952	4,200	.56	.430	593	417		8	A	37
Tyne		1955	4,470		.432	488	536	.40	13	A + A	40
Gazelle		1955	1,260	.70N	.555	226	215			A	
Artouste		1951	594	.65N	.535	248	123		5.5	A + C	8.8
Astazou		1956	355	.727	.682	198	62			A + C	
Bastan		1957	741	.67N	.535	309	146		5.2	A + C	
Marcudau		1954	386	1.07	.908	146	81		3.9	C	
Turmo	3	1953	786	.72	.675	150	143		5.5	A + C	10.5
Kusnetzov	K	1953	13,000	.571	.390	1145	661	.35	13.0	A	130
	NK-4	1954	4,500	.55	.685		305			A	55

BY PASS ENGINES

NAME	DESIGNATION	YEAR	F	$\frac{W_F}{F}$	$\frac{W}{F}$	$\frac{F}{A}$	$\frac{F}{L}$	C.R.	COMPRESSOR TYPE	W_a
Conway		1951	13,000	.70N	.250	1325	1211	12	A + A	
Soular		1957	700	.80	.44	381				
Turbofan	JT-3	1958	16,000	.70N				12.5	A+A	425
Aft fan	CJ805-21	1958							A	

HOT GAS GENERATOR

NAME	DESIGNATION	YEAR	GHP	$\frac{W_F}{GHP}$	$\frac{W}{GHP}$	$\frac{GHP}{A}$	$\frac{GHP}{L}$	C.R.	COMPRESSOR TYPE	W_a
Oryx		1954	950	.62	.526	450	136	6.1	A	15.2

** Super charged turboprop - constant power up to 20,000 feet

Navy cognizance and 10 under Air Force cognizance. Actually only about half of these (J-54, J-75, J-79, J-83, J-85, J-87, J-91, and J-93) are in an active state of development and production. The T series went to 49 and 50 in the first report, and now runs to 61 on odd numbers, and 66 on even numbers. Of these 14 new considerations about three fourths of them (T-51, T-53, T-54, T-55, T-56, T-58, T-60, T-61, T-62, T-64, and T-66) are still under development or production. Thus it is apparent that more relative emphasis has been placed on turboprops, largely from the desire to develop gas-turbine engines for helicopters.

Trends for Turbojets

It is interesting to plot again the achievements versus time. In spite of the scatter of the points, there exists some semblance of trends. For completeness and continuity the points given in the earlier 14-yr progress report are included here to give a picture of the progress for the entire 20 years of jet propulsion. The difference in the progress in the first 14 years and the last six can readily be seen. In addition to the points the first trends (approximation) are included again with their logical (?) extensions.

Size. The maximum size of axial-flow turbojets continued to increase at an even more rapid rate than during the first period, Fig. 24. Now there are turbojet engines with static thrusts of over 20,000 lb without afterburning; whereas first-period maximums were all less than 10,000 lb. Apparently the only upper limit on the size of such engines is the fabrication limit. Engines with airflows of 400 lb per sec are being considered. At the other extreme, the very small thrust engines were again designed. The thrust of the only large centrifugal-type turbojet exceeded the previously suggested trend line. However, here there appears to be an upper bound on size. A completely new trend line for axial flow turbojets has been made starting in 1946.

Specific Fuel Consumption. The trend on specific fuel consumption has continued downward throughout this period, Fig. 25, and it did not flatten out as quickly as originally indicated. Some of the latter points, however, are for normal power instead of maximum power. A partial explanation can be seen from Fig. 26, where the specific fuel consumptions are plotted versus compression ratio. This graph shows that during this period the reduced fuel consumptions were influenced more by the increased compression ratios than by improved component efficiencies.

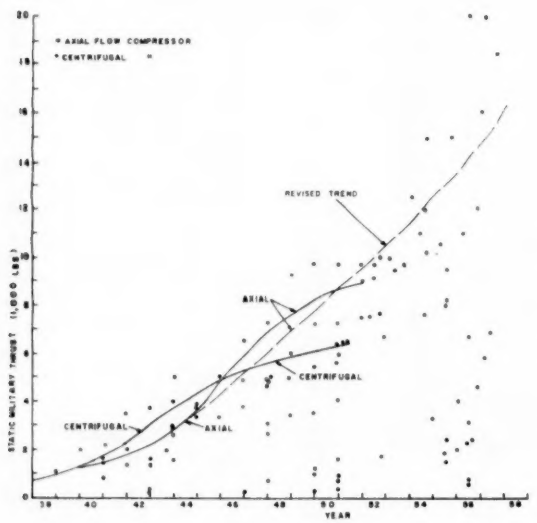


Fig. 24 Thrust of turbojets

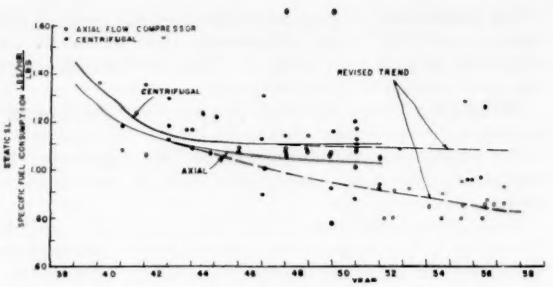


Fig. 25 Specific fuel consumption of turbojets

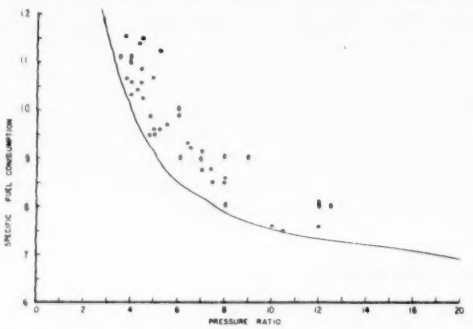


Fig. 26 Specific fuel consumption of turbojets versus pressure ratio

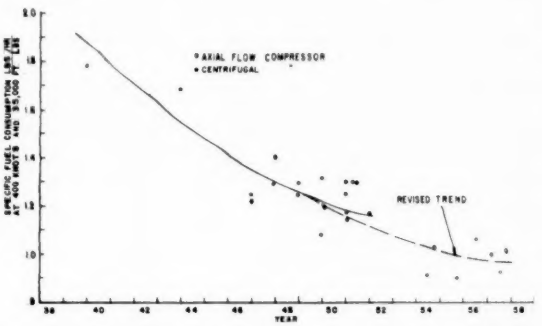


Fig. 27 Specific fuel consumption of turbojets at 400 knots and 35,000 ft

The trend in the specific fuel consumption at flight altitude and speed has also decreased more rapidly in the last few years, Fig. 27.

The specific fuel consumption with afterburning, Fig. 28, has maintained a downward trend but not as sharp as that for the basic engine, and practically no data are available on the liquid consumption for injection.

Specific Weight. Fig. 29 shows that the optimistic specific weights predicted by the British in the last report are being approached by regular engines, and have been exceeded by the expendable-type engines. It doesn't appear that the trend can go much lower than the present 0.25 yet again this may be too conservative. It is interesting to note that some of the axial-flow engines have lower specific weights than any of the centrifugal turbojets (however, not at the same date).

Thrust per Unit Size. The thrust per frontal area, as shown in Fig. 30, has continued up at a rapid rate to as much as 1450 psf in production engines, and as high as 1800 psf in tests. Here the trend line in the first report was reasonably correct for the axial-

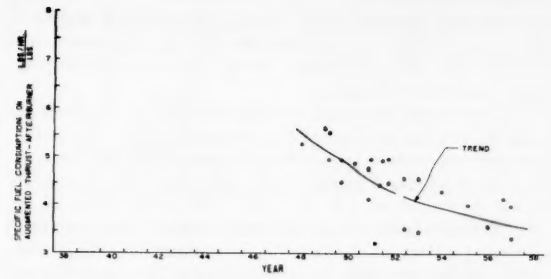


Fig. 28 Specific fuel consumption on augmented thrust—afterburner

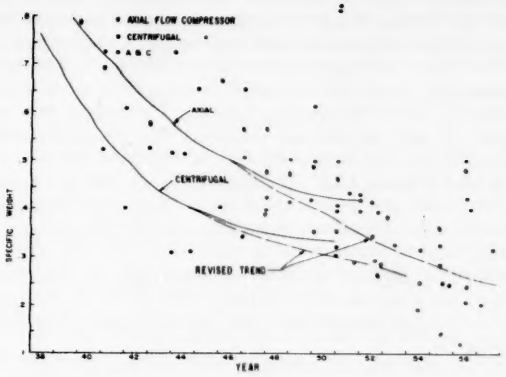


Fig. 29 Specific weight of turbojets

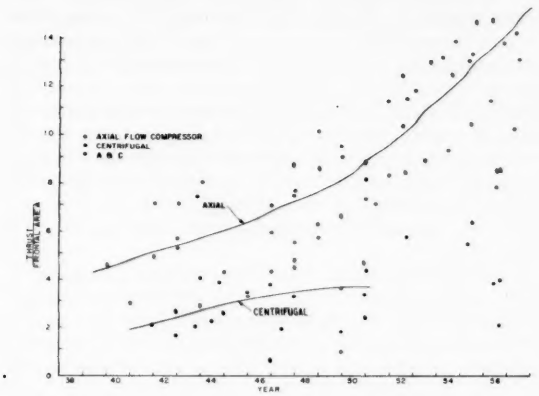


Fig. 30 Ratio of thrust to frontal area for turbojets

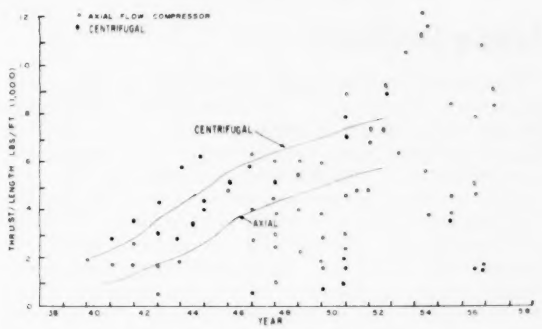


Fig. 31 Ratio of thrust to length turbojets

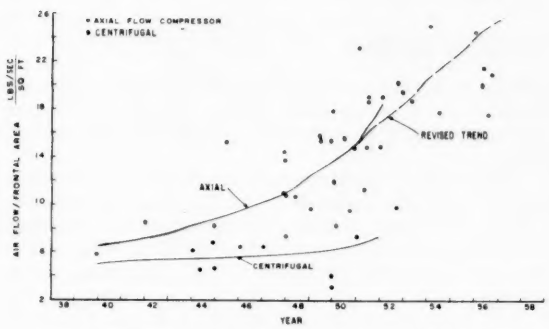


Fig. 32 Ratio of air flow to frontal area turbojets

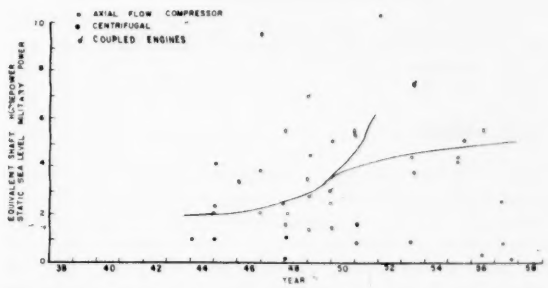


Fig. 33 Equivalent shaft horsepower for turboprops static sea level military power

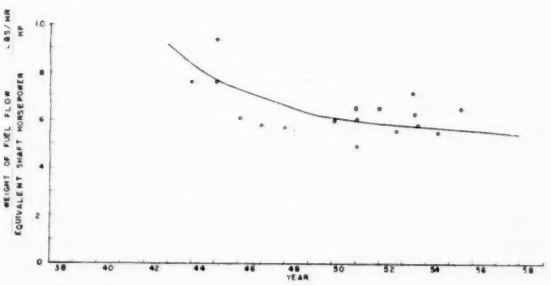
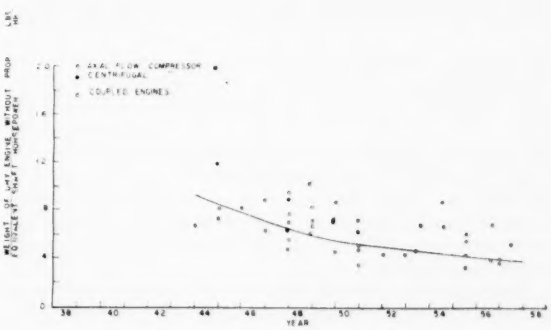


Fig. 34 Specific fuel consumption of turboprops static sea level military power



Figs. 35 Specific weight of turboprops

flow engines. On the one centrifugal-compressor engine, the thrust per frontal area shows a marked improvement up to 586 lb/ft, a 30 per cent improvement over the best in the first period. The thrust per unit length increased to over 1000 lb/ft. The rapid increase with thrust per unit length was largely a result of the increase in size of the engines, Fig. 31. The lengths were not shorter, in fact, the converse was true. The foregoing merely shows that the lengths need not be proportional to thrust.

Air Flow to Frontal Area. The increases in air flow per square foot of frontal area were of the order of 10 per cent. Since the value for none of these new engines exceeds the value of the improved Sapphire, the limit in this direction may be near. This is probably not the case, however, for when only the compressor area is considered, test engines have swallowed 30 lb of air per sec per sq ft, and some research units have gone up as high as 33, see Fig. 32.

At the same time the airflow per unit area was increased, the thrust per pound of air was increased from the range of 55–60 lb per lb per sec to a range of 60–65.

Turboprop Characteristics

The trends of the fundamental characteristics of turboprops are given in Figs. 33 through 35. Except for the one Russian design, the new turboprops for airplanes have tended to be about the same size or smaller than those developed in the first period, while those developed for helicopters are in general much smaller.

There has been a definite trend downward in specific fuel consumption both at military power and for the cruise. The former is shown in Fig. 34. The latter has decreased from about 0.6 to 0.4 with some values lower. Fig. 35 shows that progress has continually reduced the specific weights of the turboprops but the trend has not been as pronounced as for the turbojet when the sea-level static condition alone is considered. At flight speed and altitude the supercharged engine shows a 27 per cent improvement over engines of the preceding period.

The power per frontal area and per unit length of the new turboprops where comparable were essentially the same as they were for the preceding period.

Cycle Changes

The compression ratios of the subsonic turbojets and of turboprops have continued their rapid rate of increase to values as high as 13 to 1, Fig. 36, made possible by the two-spool and the variable-angle stator-blade designs. The compression ratios of the single-spool engines, without variable-angle blades, have remained below nine. The compression ratio of one of the two variable-angle stator-blade engines is 12, and apparently engines of this type could be successful with much higher values. Because of the high ram pressure, high mechanical-compression ratios are not required for engines designed for supersonic flight velocities, so some of the new engines have compression ratios of six or less. Consequently they have been of the single-spool type. There was considerable research on transonic compressor stages, those in which the velocity is supersonic over portions of the blades. For a time it was thought such stages would be the answer to the problem of obtaining high compression ratios per stage with associated lightweight compressors. This might be the case in the future, but at the present time the compression ratios per stage, Fig. 37, are no larger than they were for the better compressor of the first period. No current engines employ a supersonic compressor.

The maximum polytropic compressor efficiencies showed practically no improvement, remaining at about 0.89 to 0.90. The concentration of effort was directed to getting high efficiency over as much of the compressor map as possible, rather than increasing the maximum value at a single point. In fairness, it should be mentioned that high pressure rises per stage of 1.17 or more were maintained to high compression ratios, without a drop in polytropic compressor efficiency.

The maximum polytropic turbine efficiencies, on the other hand, have made substantial improvements during this period. Some companies have been able to improve their polytropic efficiencies as much as 6 points, thereby obtaining over-all adiabatic turbine efficiencies of 0.91 per cent. The continuation of the trend to more turbine stages has been a factor in these improved efficiencies. However, it cannot be the main explanation, for the power removed per stage has remained high. It must be the result of better flow designs.

One of the most discouraging aspects of this period was the small increase in maximum turbine-inlet temperatures. The maximum operating temperatures have remained between 2100 and 2200 R; however, at least in tests, these temperatures have been admitted for uncooled blades. Thus with cooling it appears that the maximum admissible temperatures have increased by about 200 R. The points in Fig. 38 are for uncooled blades and are based upon the results of one source. Hence they may be more significant than the previous maze of unrelated points, some cooled and some uncooled. This increase has resulted more from new stressing techniques than from the development of new metals. No basically new high-temperature metals came into use during this period.

A great deal of progress has been made in the designing of blades and blade liners to improve the potential of air cooling of the turbine blades. Yet there was no trend toward more cooling of blades, if anything, the reverse was true. There is still discussion about liquid-cooled blades, but no engines employ this principle. A number of ceramic or mixed materials are being tried for possible use as uncooled blades. Some of the latest are silicon carbide, chrome aluminum, and titanium carbide mixed with cobalt or nickel.

Combustion and Fuels

A new type of combustor appeared during this period called the cannular. As its name implies it is a combination of the can type and the annular type used exclusively during the first period.

Studies in flame structures and intensities led to the development of more intensive burning which in turn has permitted an increase in the blowout and the operating altitudes. Although the burner efficiency drops with the altitude, the efficiencies of most main burners are from 98 to 99 per cent in the normal operating range.

Considerable effort has been devoted to the search for high-energy fuels which has taken two approaches: (*a*) The solution or suspension of substances with high heats of combustion in the regular hydrocarbon fuels, and (*b*) the utilization of completely new fuels. The NACA has been quite successful in increasing the Btu per lb of fuel several percentage points by using slurries of metals in regular fuel.

The search for completely new fuels was approached by a careful consideration of all elements and their possible compounds. From such a study it appeared that the hydrogen compounds of boron were the most promising. Beryllium was better from a pure energy standpoint, but its low availability and extremely toxic gases made it less desirable. When the borons were first considered, the amount of boron in the earth's crust was not known, but it was soon found to be ample. Although the energy per unit volume of fuel is important, the most significant factor is the energy per unit weight. The approximate heating values of some of these compounds are as follows:

Beryllium hydride	32,000 Btu per lb
Diborane	32,000
Pentaborane	30,000
Tetraborane	30,000
Decaborane	28,000
Methylborane	26,000
Ethyldiborane	25,000
Tetraethyldiborane	21,000

When we remember that the heating value of the usual hydrocarbon fuels is only about 18,700 Btu per lb, a glance at the values listed gives some idea of the potential. The words "idea of the potential" were used advisedly, because the true potential is much greater than the ratio of the heat contents. In an airplane the addition of a single pound of fuel multiplies to several pounds when converted to aircraft gross weight. When fewer pounds of fuel are required, the wings, the fuselage, the engine, in fact, all parts of the aircraft may be made smaller. There is not a definite aircraft size for any mission, regardless of the energy content of the fuel. The gross weight of some of our current military planes consists of about 50 per cent fuel weight. An increase of 50 per cent in Btu per lb of fuel could cut take-off gross weight more than half, and this weight reduction would lower total Btu requirements by 50 per cent. This in turn reduces the fuel required to a fourth, thus minimizing supply and other auxiliary problems.

The boron fuels have excellent burning characteristics, and would admit a shorter combustion chamber; the efficiency would be excellent beyond our aerodynamic ceilings. The specific weight of the pentaborane and some of the other compounds is quite satisfactory. There is, however, one very serious objection from the standpoint of gas-turbine uses. The boron oxides are liquids or solids at the current temperatures of the gases passing through the turbine. The solid or liquid particles will collect on the turbine blades reducing their efficiency to a low point, thus preventing the potential saving and at the same time reducing the overhaul intervals and the engine life. In spite of this adverse characteristic, the Defense Department is spending millions of dollars in developing commercial plants for large-scale production. This one characteristic will undoubtedly prevent the use of these pure hydrogen boron compounds as the sole primary fuel for gas turbines until metals have been discovered, or some other method developed which will permit much higher turbine-outlet temperatures. Of course these fuels could be used in afterburners, and in this manner they would enhance the possibilities of high-speed flights which require sustained use of afterburners.

Nozzle Changes

During this period the exhaust nozzle ceased being just a simple contracting piece of pipe. (Although the very first jet engines had variable-area nozzles the trend then was to omit them.) Variable areas became more and more important as the higher and higher speed encouraged more and more afterburning. Not only the area but type change now. Converged nozzles are not sufficient. De Laval convergent-divergent area is desirable at some of the operating conditions. The problem of designing a leak-proof variable nozzle is a difficult one. The external devices for varying the area are known under the names of letter boxes, eyelids, calipers, clam shells, and iris. The iris is one of the best of these since openings are always circular, concentric, and conveniently cooled. One of these is shown in Fig. 8 on the afterburner of the J-79. When these are at the end of an afterburner a blanket of cool air should be provided to avoid thermal distortions. The external cooling problem would be much simpler if the area were varied by a movable center cone as in the first German turbojet. However, at present the high temperatures in the center of the nozzle have precluded this type.

The French introduced the novel idea of aerodynamically controlling the nozzle shape. They found that the effective area could be controlled by radial flow of compressed air injected at the jet-exit section.

As early as 1945, people became interested in spoiling and reversing thrust in order to decelerate aircraft. They also wanted thrust deflectors to give a component of lift for take-off and landing. During this period successful thrust reversers were constructed. One type successfully tested by Westinghouse had a series of circular turning vanes covered by a sliding shield. When the shield was moved to uncover the vanes the high-pressure gases escaped and were directed forward, thus reversing the thrust. Rolls Royce accomplished this by using eyelid shutters over normal jet-exhaust orifices to deflect the gases through airfoil vanes.

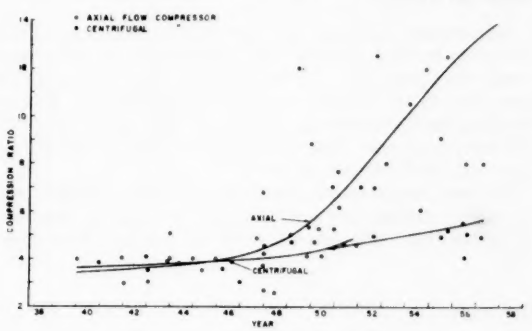


Fig. 36 Compression ratio of turbojets

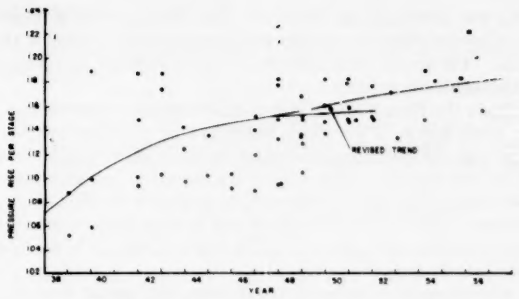


Fig. 37 Pressure rise per stage axial flow compressors

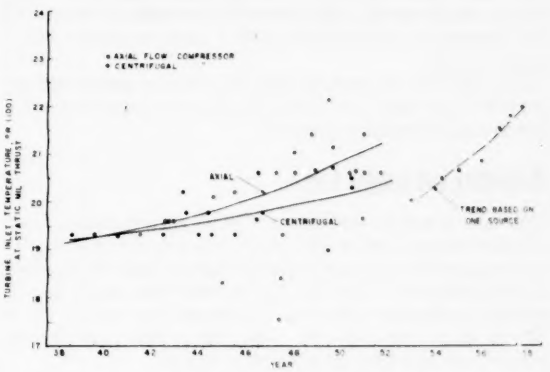


Fig. 38 Turbine inlet temperature of turbojets

This was developed for the Avon. The NACA tested a target-type hemispherical thrust reverser and a truncated version of the same. The maximum reverse-thrust ratios obtained were 59 and 52 per cent, respectively.

Again the French SNECMA used aerodynamic controls to reverse the thrust. They used a radial flow jet of high-pressure air bled from the compressor to deflect the main jet through a series of annular deflector vanes outside the normal jet stream. Complete thrust reversal is obtained in a matter of seconds by this method. This reverser was first tested in 1952 on the Goblin engine in the Vampire and the landing run was reduced to one half the usual distance.

Another major problem attacked during this period was noise. The use of deflector teeth on the nozzle prepared by Westley was the first successful device. This led to the development of the Rolls Royce pedal nozzle which gave a marked reduction in noise without loss in thrust. The Curtiss-Wright Zephyr, a version of the Sapphire, is equipped with both a sound suppresser and a thrust reverser.

In addition to the work on noise reduction on specific engines, much has been done to produce a movable noise suppresser which can be used on airplanes at airports.

Reliability and Engine Life

Reliability and life are important on an aircraft engine, where a failure means a loss of life. There was a tendency at first to overhaul and inspect engines more frequently than was required. During this period statistical studies were made which showed that the overhaul periods could be increased from the low 150 to 200 hr up to over 1000 hr. The engines have now demonstrated high reliability and sufficiently long life.

Testing

In order to adequately develop the information necessary to bring about the refinements reported here, the test facilities have grown considerably. The first turbojets were tested in simple cells with measurements confined to thrust, fuel consumption, rotational speed, and air flow. The performance at altitude was estimated from these tests until units could be flight tested. There are now several elaborate large test facilities, which enable the testing of large complete units under simulated altitude and flight conditions. Refrigeration and evacuating equipment has been made to supply the tremendous quantities of air handled by these engines. Costly instrumentation permits intricate measurements of the nature of flows, pressures, temperatures, stresses, vibrations, tip clearness, accelerations, combustion intensity, ignition, and many other things. The basic measurements are fed directly into data-processing machines and many of the results are analyzed automatically with curves being plotted mechanically.

In the early flight tests the instrumentation was quite limited and the data consisted of what the pilot could jot down. Later photographs were taken of the more elaborate instrument panels. Next, the data were transmitted directly to the ground by telemetering. Currently, these data are recorded on magnetic tape, so that it too can be fed directly into electronic computers, whereby hundreds of instruments may be recorded simultaneously in a form convenient for interpretation.

Tests on complete units may be divided into: (*a*) Development tests, (*b*) type or endurance tests, (*c*) flight tests, and (*d*) acceptance tests. Limited use of the last three types of complete-unit tests was made in the first period, but component testing has increased immeasurably. It has been reported that one manufacturer utilized 75 production cells, costing $5 million for jet-accessory production testing with temperatures from −80 to +1000 F, and simulated altitudes up to 75,000 ft. Others simulate altitudes up to 100,000 ft.

Controls

Aircraft gas turbines have many more requirements than other applications, operating in all orientations, at velocities from zero to hundreds of miles per hour, from altitudes of zero to thousands of feet, with entering air temperatures from −70 to +600 F; and at these conditions they must operate anywhere from idle to full power.

The engine controls on aircraft gas turbines should insure that the engine will operate properly under all of these conditions and in addition should: (*a*) Accelerate the engine from idle to full power in less than 10 sec at sea level, (*b*) give the least fuel consumption for a given power at all operating conditions, (*c*) protect the engine at all times, and (*d*) protect the pilot and engine in case of engine failure.

Moreover, in order to relieve the pilot, this should all be accomplished with one control lever.

The controls must protect the engine from the usual gas-turbine physical limits of maximum temperature, maximum rotational speed, and compressor surge common to all gas turbines. In addition, they must give protection against burner blowout, excessive flame lengths (at low densities), and high intake pressures (at high Mach numbers).

Not all of the requirements have been fulfilled, but great strides have been made. On acceleration, the pilot may open the throttle and the controls will prevent the engine from encountering the surge line by changing the guide vanes as in the J-79, the air bleed as in the J-54, or by varying the nozzle areas so as to permit the rpm and corresponding air flow to increase before the compression ratio and temperature increase beyond their limit. These controls are base-pressure and temperature-sensing instruments, with over-riding devices which correct for air pressures and temperatures. On rapid accelerations there is a tendency to overshoot the mark. Feedbacks insure stable operation, after accelerations or decelerations, by damping out the oscillatory effects. Now, with most engines, a "slam" acceleration may be made without danger of surge or unstable operation.

As yet, no control insures the minimum fuel consumption at all altitudes and speed.

In the case of failure of a turboprop, the excessive wind-milling drag would be catastrophic, for the negative thrust might be so great as to throw the airplane out of control. Within the controls there are now devices which feather the propeller or coarsen the pitch upon sensing negative torque.

This and the preceding topic are so large that separate papers should be given rather than cursory mention.

Impact on Business

Aviation gas turbines are now made by the thousands. There were over 36,500 J-47's produced. Allison Division alone produced over 16,000 J-33's and 14,000 J-35's. Over 2000 Sapphires were made in the United Kingdom, and over 8000 by Curtiss-Wright in the United States. By March, 1957, over 8000 J-57's had been produced. 1000 Derwents have been built in Belgium and 550 Verdons in France.

The total employment in the United States in connection with the design, testing, and production of these engines was over 120,000 in 1957. This was more than for any other year except 1953.

Summary

In the years from 1952 to 1958, the gas-turbine became an in-

tegral part of the aircraft industries. In fact, the gas turbine has completely revolutionized the industry both in thinking and in designs. Research and testing have been stimulated more than by any previous development. Practicability for commercial use, both as turbojets and turboprops, has been demonstrated, and use in helicopters has begun. Aircraft have been propelled at Mach numbers greater than 2.

The improvements in the engines have been more in the nature of refinements. Specific weights have been reduced to as low as 0.25, specific fuel consumption to 0.76 lb per lb of thrust, and 0.41 lb per eshp. The time between overhaul has been doubled. The centrifugal-type compressor has given way to the axial-flow type.

The two most important innovations were the two-spool units and the variable guide vanes. The most novel developments were the bypass engine and a supercharged turboprop.

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ROCKET TURBINES

Rocket turbines have come into prominence in the last 20 years chiefly as sources of power for rocket pumps and auxiliary missile requirements. Most are of an impulse type with low efficiencies yet very low weights. Varying from 2 to 20 in. in diam with powers from 1 to 15,000 hp, they are designed to operate for durations of 1 to 5 min with total life of 1 to 50 hr. Although rotational speeds are as high as 100,000 rpm, the tip Mach numbers are lower than in conventional turbojet blades.

Prior to 25 years ago the rocket turbine was rarely mentioned, but since that time it has grown to be of great importance, especially in the missile field. Here the term "rocket turbine" is defined to be a turbine driven by rocket exhaust gases. Perhaps the forerunners of these turbines were the gas turbines employed to drive torpedoes. Although practical working liquid rockets were claimed [12] to be operated by a Peruvian as early as 1895, the first authenticated shot of a liquid-fueled rocket was by Robert H. Goddard on March 16, 1926, at Auburn, Mass. The liquid fuels of these early engines were fed to the combustion chamber by pressurized or intermittent injection systems. It is not generally known who first conceived the *idea* of employing a gas turbine to drive the fuel pumps. Some think it was Goddard, others Oberth, others Sanger, and it may have been Ziolkowsky. We do know actual experimental work was begun in 1932–1934 by Dr. Goddard.

Applications

One of the earliest uses of the rocket turbine was in the Walter 109-509 rocket, the power plant for the first operational rocket-powered airplane, the German ME163C. Following this was its use in the well-known German V-2 rocket. In each of these a turbine drove two pumps, which supplied the oxidizer and the fuel to the main combustion chambers. Today all large and all long-duration rockets use a turbopump fuel system. Fig. 1 gives a rough estimate on the boundary of rocket size and duration for use of pressurized fuel systems and turbopump fuel systems.

In modern rockets, such as the Vanguard, the turbines are employed to supply auxiliary power, in addition to pumping the fuel. For other military applications, special units have been developed for auxiliary power sources. They drive coolant pumps, hydraulic pumps for the movement of controls and various other purposes, and small generators to provide electrical power for guidance and control activation, Figs. 2 and 3.

These turbines range in power from 1 to 15,000 hp. When one considers the size of the V-2 rocket and its turbopump, he would conclude that the 15,000-hp turbine is large enough to supply fuel to a rocket which would produce 1,600,000 lb of thrust.

Another very promising application is found in a recent hybrid engine—the air-turborocket, Fig. 4. This engine consists essentially of a modified turbojet in which the turbine that powers the compressor is partially driven by the exhaust products of a rocket. A secondary stream of air passes through the compressor and mixes with turbine-exhaust gases, where the oxidization of rocket fuels is completed, and other fuel may be burned. Ranked between the turbojet and the rocket, this hybrid engine is more independent of altitude and speed than the turbojet, and it consumes less fuel than the rocket. The internal rocket acts as a pilot burner, thus there would be some combustion at all altitudes.

By far the most important application for rocket turbines is still that for driving the propellant pumps. There are three basic turbopump systems; the bleed, the topping, and the separate gas-generator unit. Each of these systems is illustrated in Fig. 5. In the separate gas-generator system there is a small distinct rocket system constructed for the sole purpose of producing the gases to run the turbopump. The gases are created in the gas generator and after they impinge on the turbine blades they are expanded to ambient conditions through a nozzle. In the bleed system, the gases which drive the turbine are piped from the main combustion chamber directly to the turbine nozzles after which they are expanded to ambient conditions. In the topping system, the turbine-exhaust gases are injected into the main combustion chamber. In either the gas-generator unit or the topping system, the propellants for the gas generator may be supplied by a separate source or may be obtained directly from the pump lines which carry the fuel and oxidizer to the main combustion chamber. In the latter case, as well as with the bleed system, a starting mechanism is required. A fourth system, the blast turbine, in which the turbine operates in the hot exhaust gases of the main motor, has not met with success.

Currently the most troublesome problems in the design of a rocket-turbopump fuel system are not the turbine components but the seals, the bearing, and cavitation. However, discussion will be focused upon the turbine.

Types of Turbines

Because of their simplicity, their low weight per horsepower, and their ability to operate at high speed and high temperatures for short intervals of time, the impulse turbines are used almost exclusively for rocket applications. On the very large rockets, these impulse turbines are multiple-stage, Curtis velocity-stage turbines, or pressure-stage turbines, ranging from 2 to 6 stages, Fig. 6. On the small rockets, the simple single-stage impulse turbine is by far the most frequently used type. Moreover, in order to obtain higher specific speed at low powers, many of these have only partial admission. That is, the turbine nozzles supply gas to only a portion of the turbine blades, Figs. 7 and 8. For a fixed diameter and a fixed rotational speed, the power delivered is a function of the admission. Currently, rocket turbines employed to drive propellant pumps for rockets will have full admission when the thrust of the main rocket is large, say, over 10,000 lb.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, August 26, 1958. Paper No. 58—A-46H.

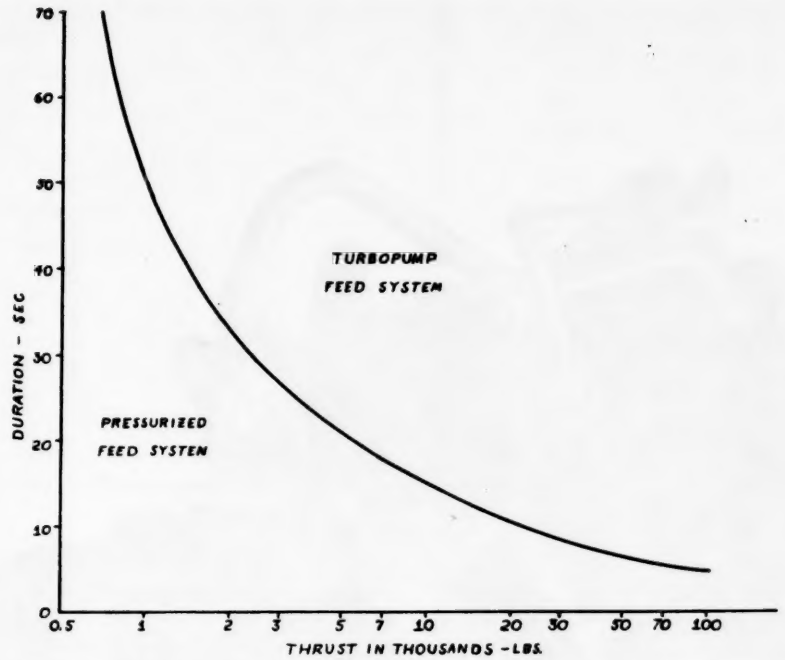


Fig. 1 Approximate boundary on thrust and duration of a rocket for use of turbopump fuel system

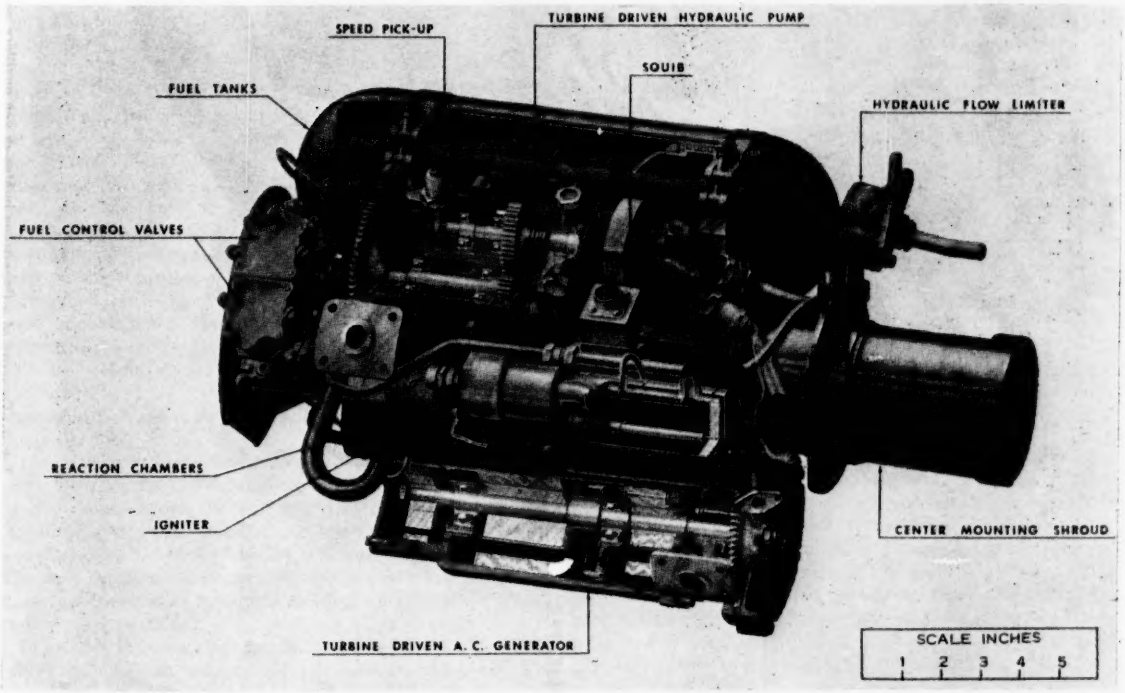


Fig. 2 Cutaway of single-stage auxiliary-power supply package (TU132000) (Courtesy Thompson Products, Inc.)

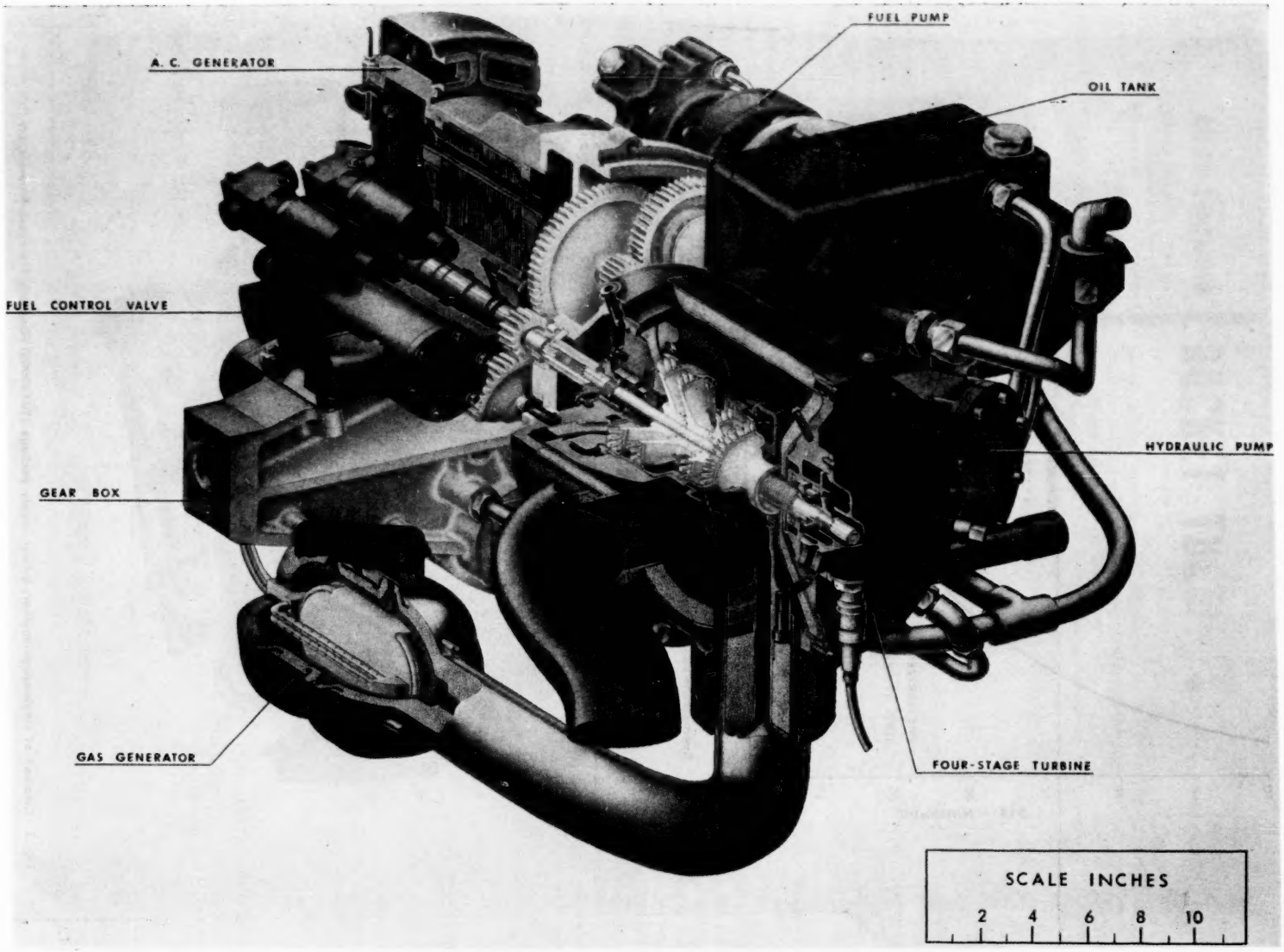
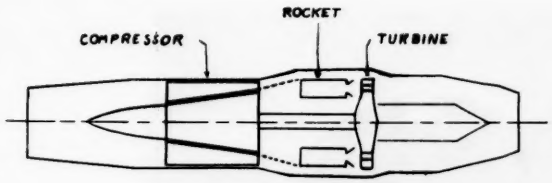


Fig. 3 Cutaway of a three-stage turbine-driven auxiliary power supply (TU-112700) (Courtesy Thompson Products, Inc.)



AIR - TURBOROCKET

Fig. 4 Sketch of air-turborocket engine

The German Walter 109-509A rocket engine had a single-stage re-entry turbine. There was only partial admission of the high-pressure gases, then there was a second re-entry of the low-pressure gases. The nozzles were distributed along the periphery of the turbine casing, so that the gases were admitted only at certain parts. After flowing through the moving turbine blades, the gases were then collected and directed to re-enter the turbine wheel at other points, thereby supplying additional energy to the turbine. In this country, Aerojet has produced a three-pass re-entry turbine.

Turbine Efficiency

The efficiency of a simple impulse turbine, as given in most texts, is a function of the ratio of the mean chord circumferential velocity to the nozzle-gas velocity, Fig. 9. For a maximum efficiency of a single-stage turbine, the rotational tip speed should be about one half the nozzle jet velocity. The turbine wheel could run this fast, but unfortunately there would be cavitation in the pumps, with a reduction in efficiency of the complete process. Consequently the turbines are designed to run at low speed. As a result, the over-all turbine efficiencies range from 12 to 75 per cent with many of them operating at about 40 per cent. These low efficiencies are acceptable in certain missile applications. In fact, for some purposes, the efficiency is sacrificed for reliability. It is more important that the turbine pump function than it is that it function efficiently. Hence blade-tip clearances are made sufficiently large to insure operation under extreme conditions (note first stage in Fig. 10). Since the energy required to run the turbopump is only about $1^1/_2$ to 3 per cent of the total employed in the mission, the total waste in fuel is not great for small missiles.

In long-duration operations such as would be required for aircraft propulsion or in the establishment of space vehicles where the pay load is critical (to establish escape velocity it takes about 1000 lb of weight per each lb of pay load), these low efficiencies are not satisfactory, for a small percentage of the extremely large amount of fuel would be equivalent to a many-fold increase in the payload.

Increases in turbine efficiency can be accomplished in many ways, but all are at the expense of more hardware weight. The problem is to minimize the total weight of the fuel and metal for the mission. Partial admission and low velocity ratios lead to low efficiencies. Partial admission is no problem on large units. The more common way to improve the turbine efficiencies is by increasing the relative rotational speed of the impulse turbines, and by pressure staging.

In general the rotational speeds of the impulse turbines are below the optimum velocity ratio. More favorable velocity ratios have been accomplished in many designs by inserting a gearbox which permits the turbine to operate at a higher speed than the fuel pump. Another design incorporated the two-spool concept with a two-stage pump, where the low-pressure pump stage was driven through a gear train at a lower rotational speed while the high-pressure stage was driven directly by the main shaft at

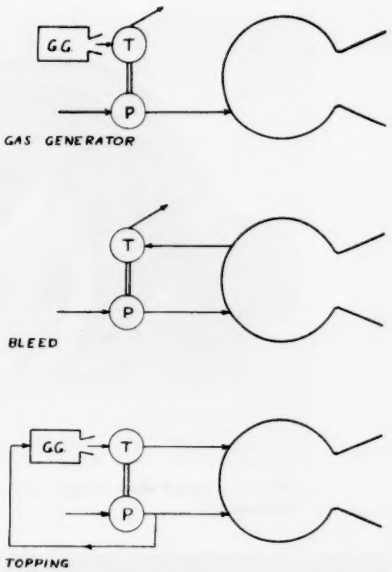


Fig. 5 Diagram of three turbopump systems; the topping, the bleed, and the separate gas-generator unit

the full turbine speed. This produced an efficient system and at the same time permitted an over-all reduction in weight over a system where all of the power passes through a gearbox. It is believed the Russians used a two-stage pump in their Sputnik motors, where one stage pumped the propellants for cooling, then the other stage pumped them into the combustion chamber.

It may seem strange for us to say that rotational speeds are too low for best efficiency when you realize that the rpm are as high as 100,000 for very-small-diameter turbines, and most have speeds between 20,000 and 35,000 rpm. In spite of this, the tip speeds are less in general than they are in the conventional gas turbine. The diameter of the turbine with 100,000 rpm is less than 2 in. In general, the tip Mach numbers of the rotors are much less than 1, yet the Mach number of the rocket gases, relative to the rotor, may be up to about 2.

In the topping turbine system, mentioned previously more favorable velocity ratios and higher turbine efficiencies are obtained by reducing jet velocities instead of increasing the rotational speed. A topping turbine operates under a low pressure ratio; viz, the ratio of the gas-generator pressure to the combustion-chamber pressure. Hence the turbine can operate at a relatively high efficiency at comparatively low rotational speed—speed low enough for direct-drive operation.

With the high pressure ratios, some improvement in efficiency and more power can be developed by pressure staging. In these turbines the pressure energy is converted into velocity energy in a series of stator stages, each followed by an impulse rotor. Better efficiencies could be obtained by employing reaction-type turbines, those in which there is an expansion or pressure drop in the turbine rotor as well as in the stator. One manufacturer has designed a turbine with a large blade-height-to-diameter ratio in which the root sections are of the impulse type and the sections gradually change to the reaction type at the tip.

Fuels

A variety of liquid propellants and some solid propellants have been used for the gas generator. Perhaps the most popular of these is concentrated hydrogen peroxide with a permanganate or

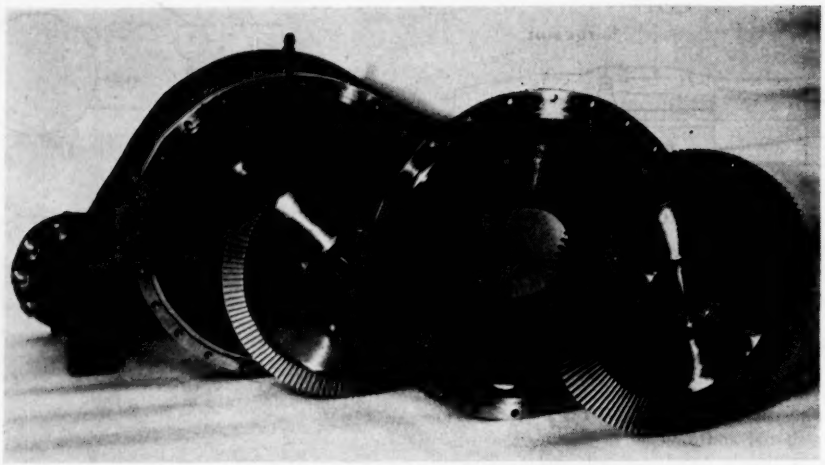


Fig. 6 Exploded view of major components of a two-stage rocket turbine (Courtesy Aerojet-General Corporation)

Fig. 7 Cutaway of turbine manifold showing partial admission of nozzles to turbine wheel (Courtesy Bell Aircraft Corporation)

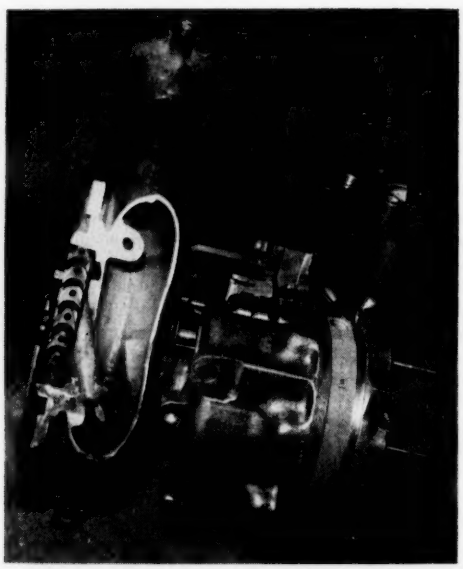


Fig. 8 Cutaway of turbine wheel and manifold showing a nozzle and turbine blades (Courtesy Bell Aircraft Corporation)

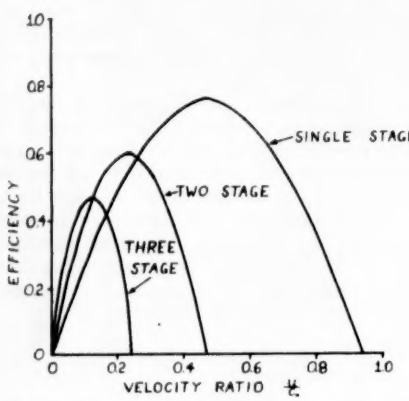


Fig. 9 Efficiency of single and multiple-stage impulse turbines as a function of the ratio of the mean chord circumferential velocity to the nozzle-gas velocity

silver-screen catalyst. Other fuels include: Liquid oxygen, LOX, and kerosene; LOX and alcohol; LOX and ammonia; LOX and JP4; red-fuming nitric acid, IRFNA, and kerosene; red-fuming nitric acid and mixed-amine fuel, MAF; white-fuming nitric acid, WFNA, and kerosene; white-fuming nitric acid and unsymmetrical dimethyl hydrazine, UDMH, with a diluent of water and alcohol; ethylene oxide; N-propyl nitrate; and hydrazine.

Gas-Generator Pressures

Most of the gas generators operate at a chamber pressure in the range of 300 to 800 psia. However, some few exceptions operate at as low as 85 psia and others as high as 1500 psia. Those which drive single-entry single-stage impulse turbines usually have a pressure of about 300 psia or a pressure ratio of 20 to 1. The higher pressures are usually combined with pressure-stage or re-entry turbines.

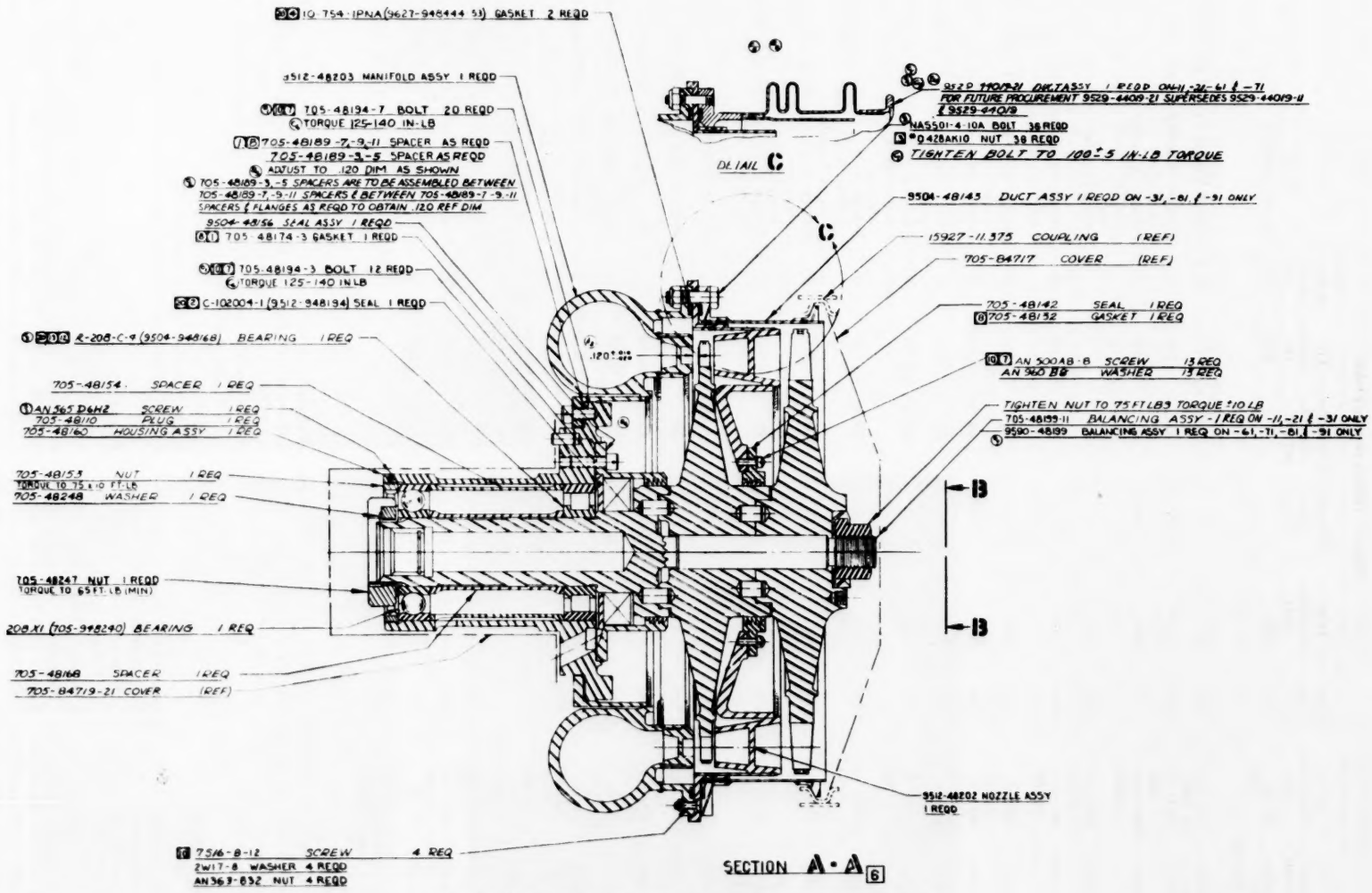

10 754-1PNA(9627-948444 53) GASKET 2 REQD
9512-48203 MANIFOLD ASSY 1 REQD
705-48194-7 BOLT 20 REQD
TORQUE 125-140 IN-LB
705-48189-7,-9,-11 SPACER AS REQD
705-48189-3,-5 SPACER AS REQD
ADJUST TO .120 DIM AS SHOWN
705-48189-3,-5 SPACERS ARE TO BE ASSEMBLED BETWEEN 705-48189-7,-9,-11 SPACERS & BETWEEN 705-48189-7,-9,-11 SPACERS & FLANGES AS REQD TO OBTAIN .120 REF DIM
9504-48156 SEAL ASSY 1 REQD
705 48174-3 GASKET 1 REQD
705-48194-3 BOLT 12 REQD
TORQUE 125-140 IN-LB
C-102004-1 (9512-948194) SEAL 1 REQD
R-208-C-9 (9504-948168) BEARING 1 REQ
705-48154 SPACER 1 REQ
AN 565 D6H2 SCREW 1 REQ
705-48110 PLUG 1 REQ
705-48160 HOUSING ASSY 1 REQ
705-48153 NUT 1 REQ
TORQUE TO 75 ± 10 FT-LB
705-48248 WASHER 1 REQ
705-48247 NUT 1 REQD
TORQUE TO 65 FT-LB (MIN)
208 XL (705-998240) BEARING 1 REQ
705-48168 SPACER 1 REQ
705-84719-21 COVER (REF)
7516-8-12 SCREW 4 REQ
2W17-8 WASHER 4 REQD
AN 363-832 NUT 4 REQD
9529 4409-21 DIA ASSY 1 REQD ON 11,-21,-61 & -71
FOR FUTURE PROCUREMENT 9529-44019-21 SUPERSEDES 9529-44019-11 & 9529-44019
NAS501-4-10A BOLT 36 REQD
*0428AK10 NUT 36 REQD
TIGHTEN BOLT TO 100 ± 5 IN-LB TORQUE
DETAIL C
9504-48145 DUCT ASSY 1 REQD ON -31,-81 & -91 ONLY
15927-11.375 COUPLING (REF)
705-84717 COVER (REF)
705-48142 SEAL 1 REQ
705-48152 GASKET 1 REQ
AN 500AB-8 SCREW 13 REQ
AN 960 B8 WASHER 13 REQ
TIGHTEN NUT TO 75 FT LBS TORQUE ± 10 LB
705-48159-11 BALANCING ASSY - 1 REQ ON -11,-21 & -31 ONLY
9590-48139 BALANCING ASSY 1 REQ ON -61,-71,-81 & -91 ONLY
9512-48202 NOZZLE ASSY 1 REQD
B
B
SECTION A-A

Fig. 10 Engineering drawing of a rocket turbine (Courtesy Rocketdyne)

Table 1 Turbine summary table

Turbine application	Shp and rpm	Inlet temp, deg F	Inlet pressure, psia	Velocity ratio U/C_0	Efficiency over-all, per cent	Tip speed, fps	Run duration per cycle, min	Type	Turbine gas
Rocket turbo-pump for Walter 109-509 rocket	90 @ 16500	825	398			578	4.6	1-stage re-entry	H_2O_2
Rocket turbo-pump for V2	465 @ 3800	725	375			523	1.1	2-stage Curtis	80 per cent H_2O_2
Rocket turbo-pump	1800 @ 3000	1450	375	0.22	57	750	3	2-stage velocity or Curtis	H_2O_2
Rocket turbo-pump	3601 @ 12000	1540	500	0.165	36	827		1-stage impulse partial-admission	
Rocket turbo-pump	850 @ 34500	1350	305		70			1-stage impulse	90 per cent H_2O_2
Rocket turbo-pump	48 @ 20000	500	85	0.23	65	567	2	1-stage axial impulse	UDMH-WFNA and H_2O-methanol diluent
Rocket turbo-pump	340 @ 38000	1250	350	0.32	63	1150	5	1-stage axial impulse	WFNA-kerosene
Rocket turbo-pump	600 @ 17500	1320	320	0.24	45	680	1	1-stage axial impulse	IRFNA UDMH-kerosene
Rocket turbo-pump	350 @ 18000	1250	540	0.19	37	700	10	1-stage axial impulse	WFNA kerosene
Rocket turbo-pump	4000 @ 24800	1312	502	0.25	60.8	1065	5	2-stage axial pressure-staged impulse	LOX kerosene
Rocket turbo-pump	2000 @ 26384	1300	611	0.275	59.2	1140	5	2-stage axial pressure-staged impulse	LOX kerosene
Rocket turbo-pump	15000 @ 6000	1300	600	0.11	58	470	5	5-stage axial pressure-staged impulse	LOX kerosene
Gas-generator feed turbo-pump	88 @ 35900	1300	658	0.250	45	1100	5	1-stage axial impulse	LOX kerosene
Rocket turbo-pump	350 @ 20000	625						1-stage impulse	
Air turborocket	2000 @ 27500	1800	750		50	1200		1-stage axial impulse	Ethylene oxide
Auxiliary-power unit	1.98 @ 12000	2200	250 cruise 1500 boost	0.04	12	210	5	Re-entry (Terry) 3 pass	Solid propellant
Auxiliary-power unit	27 @ 73000	2200	800	0.315	55	1600	15	1-stage axial impulse	N-propyl nitrate
Missile auxiliary	35 @ 24000	1400 1800	260-380		30-35	605	8	1-stage axial impulse	LOX-kerosene
Auxiliary power	1.34 @ 24000					33	1.5	1-stage impulse	Ethylene oxide
Auxiliary power	68000						240	4-stage axial	
Auxiliary power	1.3 @ 24000						1.5	3 pass re-entry 1-stage	Solid propellant
Rocket turbo-pump	189 @ 16000	366	40	0.41	75	618	2	1-stage axial impulse	Turbojet air bleed
Rocket turbo-pump	71 @ 20000	475	44.4	0.332	68.1	567	2	1-stage axial impulse	Turbojet air bleed
Auxiliary power for electric generator	0.17 @ 100000	600					0.25		Compressed air

The expansion is usually to about 20 to 25 psia. The gases are nozzled down before they are exhausted to the ambient condition. This back pressure permits operation of the turbine with the same pressure ratio at all altitudes.

Gas Temperatures

From the characteristics of some of the existing rockets given in Table 1, it can be seen that the inlet temperatures are low compared to those in the main rocket chamber. Instead of values of 4000 to 6000 F, they range from 2200 F down to as low as 500 F. This is brought about by the use of either low or high mixing ratios. In the case of the decomposition of the oxidizer H_2O_2 without a fuel, the temperature is about 1250 F, and the mixing ratio is infinite. In other cases the mixing ratios are low. In some cases the normal mixing ratio is employed, then a diluent of alcohol and water is used to reduce the temperature. Thus, with a few exceptions, the temperatures of the gases impinging on the rotating turbine blades is less than that in current turboprops and turbojets. The actual temperature to employ depends upon the duration of operation, the degree of admission, and the turbine-blade materials. Methods of blade cooling had not been employed, other than cooling the gases. Of course, partial admission admits higher temperature than full admission.

Power

Under the afore-mentioned operating temperatures and pressures, the energies available for conversion to shaft power range from 250 to 550 Btu per lb of gas. Resulting turbine-nozzle velocities range from 2500 to 5000 fps and perhaps slightly higher. It takes from 0.0055 to 0.0110 lb per sec of the propellant to pro-

duce a horsepower. It takes about 15 to 25 hp per 1000 lb of thrust for pumping the fuel. The exact amount varies with the type of propellant and the rocket propellant. In general, the lower horsepower would be associated with the lower pressures and the upper values with the higher pressures. From this it follows that from $1^1/_2$ to 3 per cent of the fuel weight is required to drive the turbopump.

Increased enthalpy per pound of fuel and oxidizer consumption can be obtained by using higher temperatures and pressures. From the formula for the jet velocity,

$$V_j = \left\{ \frac{2g\gamma}{\gamma - 1} \frac{RT_0}{M} \left[1 - \left(\frac{P_2}{P_0} \right)^{\frac{\gamma - 1}{\gamma}} \right] \right\}^{1/2}$$

where γ is the ratio of specific heats, R is the universal gas constant, M is the molecular weight of the combustion products, T_0 the chamber burning temperature, P_0 the chamber pressure, and P_2 the exhaust pressure, it can be seen that such increase in temperature and pressure would give higher jet velocities. The apparent gains in power by using superior fuels would be lost, however, by the reduced efficiency resulting from the lower velocity ratio. This nullifying effect on the desirability of the more exotic fuel plus the reliability of hydrogen peroxide are the two reasons for the popularity of H_2O_2 for a gas-generator fuel.

Construction and Fabrication

The turbine diameters range from 2 to 20 in. Even on the smallest of these, there are 40 or more blades; and on many of the others there are over 150 blades. In order to produce these turbine wheels at a reasonable cost, several methods of fabrication have been tried. These range all the way from the construction of the individual blades, which are mechanically fastened to the rotor disk, to the construction of the complete rotor in one piece of metal. In between these extremes is the formation of groups of blades which are welded or fused or mechanically fastened to the rotor disk. The turbine size and its application has a big influence on the method employed. Surely for the small turbines, the most promising method of fabrication is to use a profile milling machine to make the complete rotor from one solid forging, Fig. 11. Not only is this method cheaper, but it eliminates the knotty problem of attaching the blades to the disk. This has been attempted by milling, by electrical discharge, and by ultrasonic machining. Also, attempts have been made to construct the rotor by integral casting.

Individual blades and groups of blades have been cast or forged and fastened to the disk by mechanical means, Fig. 12, by high-temperature nickel brazing, by welding, by forging the disk around the blade fasteners, and by disk casting and fusing to the blade. Among the mechanical fastening methods is the familiar fir-tree root common in turbojet rotors, Fig. 13. A method of special interest for this application is the clevis coupling. This procedure works well for a two-stage impulse turbine where both rows of blades are attached to the same disk. A fourth method of construction is to weld an integral ring of cast blades to a forged disk.

The methods of manufacturing have a dominating influence upon the design and quite often dictate performance compromises. Machining of the rotor from forging of high-temperature alloys involves compromises in blade spacing and shapes.

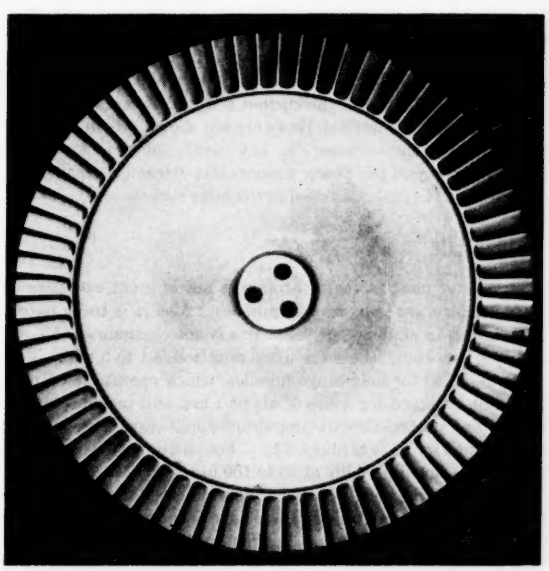


Fig. 11 7-in-diam integrally machined turbine wheel (Courtesy Aerojet-General Corporation)

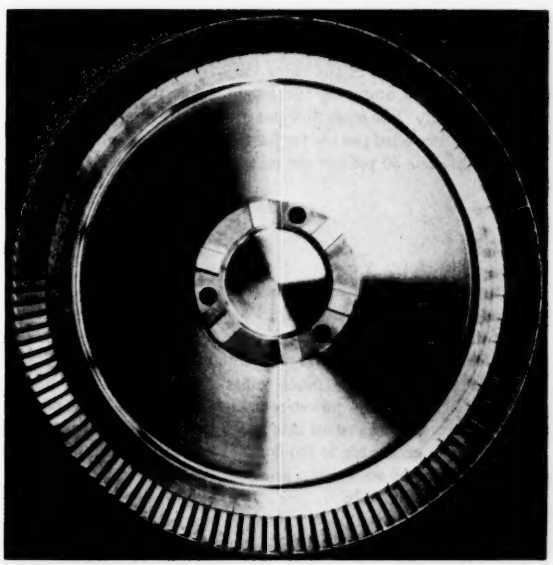


Fig. 12 10-in-diam turbine wheel after final machining showing cast-blade segments in a forged disk (Courtesy Aerojet-General Corporation)

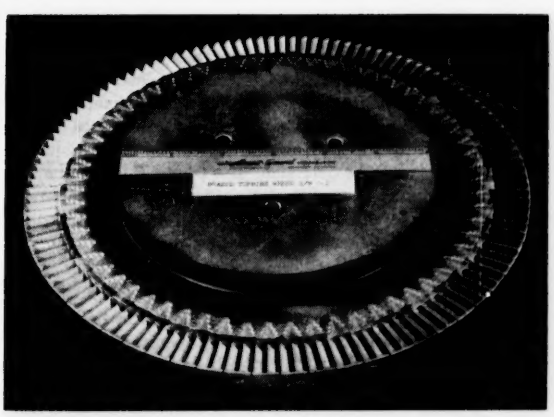


Fig. 13 10-in-diam furnace-brazed assembly showing cast fir-tree blade segments in forged disk (Courtesy Aerojet-General Corporation)

Usually only the fronts of the blades are cut in a radial direction. These modifications in shape and spacing tend to reduce the efficiency and increase the propellant consumption. On the other hand, the casting of blades permits greater design freedom but introduces quality-control and inspection difficulties.

The low operating temperatures coupled with the short periods of operation permit the use of aluminum alloys for the turbine wheels in many applications (used in V-2 for example). More frequently they are made of stainless steel and Timken, 16 Cr 25 Ni 6 Mo, although some other heat-resistant metals are used. When the blades are made separately, they have been made of Inconel X, Cast Vitallium, Stellite, Nimonic 80, and other high-temperature metals.

The hub-tip ratio of the turbine blades varies from 0.7 to almost 1.0, with the majority of the designs in the range of 0.85 to 0.91. Some of these blades are only 0.2 in. long, and most of them, in the first-stage turbine, are less than an inch. The machine depth limits the length of the blades when they are machined from a single forging. Some believe the minimum length should be 0.5 in.

The height-chord ratios are of the order of 1 in the first stages. The solidity of the rotors range from 1.25 to 1.75.

The turning angles in the rotor vary from 60 to 120 deg. In the pressure-stage units, the turning in the stator blades is of the order of 70 deg. The mass flow per over-all area perpendicular to the axis is below 5 psf per sec for partial-admission turbines, while it runs as high as 30 psf per sec on some full-admission large turbines.

Controls

Many of the rockets are controlled on the simple basis of power received equals power delivered. Power delivered by the turbine at first increases more rapidly with rotational speed than does the power for pumping but later the trends are reversed and the power required increases much more rapidly than the power available. Hence the power-required curve and the power-available curve will have an intersection at the operating point. When the gas generator is fed by fuel from its own pumps, the intersection is not so sharp; however, the system will operate without further controls.

When the rocket thrust is to be varied either by a set schedule or by an operating pilot, some form of control is necessary for the turbine so that it will supply the proper power to the pumps. With a power source arising from the decomposition of some fuel such as H_2O_2 where a liquid catalyst is required for reaction, control is easily achieved by regulating the amount of liquid catalyst allowed to flow. This procedure was employed in the V-2. Today the speed control on these turbines can have practically any accuracy desired. When low-level accuracy is required, turbine-inlet pressure regulation and mechanical flyball governors are employed to control the flow of propellants. When the required accuracy is high, electric controls based on frequency discrimination, or phase detection, or a combination of the two are used. A combination of the latter two will give an *integrated* accuracy of one part in 100,000.

Lubrication

Both oil and grease have been used for lubrication. Oil is generally favored when the drive cooling requirements are high and the duty long. When operation in hot environment is required, either a thermal lag or self-contained cooling systems are used. In some instances, the propellants themselves have been used. More frequently it is necessary to heat the lubricants rather than cool them.

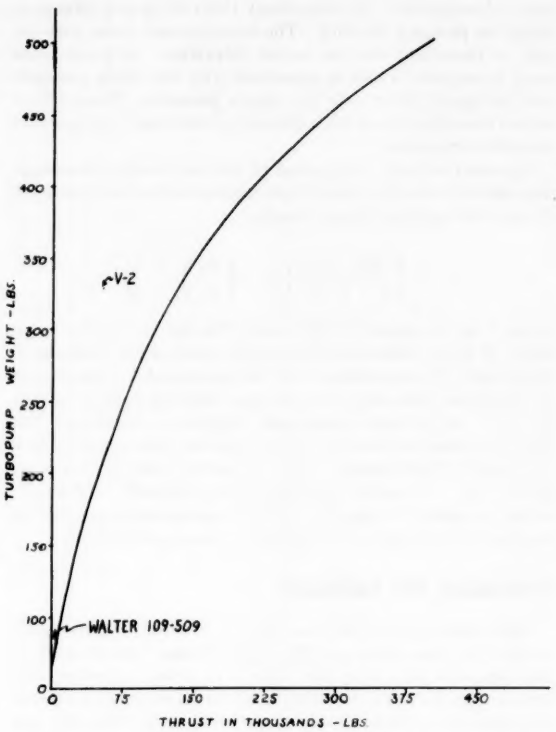


Fig. 14 Weights of turbopump systems as a function of the thrust of the rocket

Weight

As was indicated earlier, the refinements of the turbine, to a large extent, are dictated by the weight considerations. The problem is to make the over-all weight of the power plant and the total propellant consumption for the mission a minimum. Several analytic studies have been made to determine the optimum design [2]. Fig. 14 shows the weights of turbopump systems versus the thrust of the rocket. These weights more nearly represent what could be done with the current state of the art rather than the actual ones. Operation times are not shown, which would be necessary for completeness. In any event, the weight of these units is very small per power output, and extremely small when compared to the fuel consumed in the large rockets.

Turbine Life

The life of most of these turbines is not of great extent since their primary use is in various missiles. Also it is to be noted, their use as an auxiliary-power source is not continuous—15 min is a long duration; the more usual duration is 1 to 5 min. Turbines designed for long-range missiles, which operate for 5 min, would be designed for a life of about 1 hr; this includes testing time. For piloted-aircraft-propulsion application, the turbines are designed for a life of about 5 hr. For auxiliary-power sources. they are designed for a life of 50 to 100 hr.

Although earlier in this paper it was indicated that there were no troubles with the turbines, there have been some. Improperly forged billets have resulted in fractured turbine disks. This problem was eliminated by using an acid etch to check the grain of the structure. Also, cracking of the leading edges of the turbine blades at the roots has been encountered on machined tur-

bines. This was solved by reducing the mass of the blades and undercutting the turbine-blade leading edge at the root to eliminate the stress concentration at this point.

Table 1 gives a summary of leading characteristics of some of the rocket turbines. Two units operating on turbojet exhaust gases and one on compressed air are included since they are very similar to rocket turbines.

Summary

Rocket turbines have come into prominence in the last 20 years, chiefly as sources of power for rocket pumps and auxiliary missile requirements. Most of them are of an impulse type with low efficiencies yet very low weights. They exist in sizes from 2 to 20 in. in diameters with powers from 1 to 15,000 hp. Hydrogen peroxide is the most popular propellant for the gas generator; however, other conventional rocket fuels are used. In all cases the temperatures are held down by the mixing ratios or diluents. Many of the complete turbine wheels are machined from one solid forging. The blades are short with chords about equal to their length. They are designed to operate in durations of 1 to 5 min, with a total life of 1 to 50 hr. Although rotational speeds are as high as 100,000 rpm, the tip Mach numbers are lower than they are in conventional turbojet turbines.

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AUTOMOTIVE

Many experimental models of gas turbines have been built for vehicle propulsion, and indications are that production models may be available within the next decade. Special effort has been devoted in recent years to improving the fuel consumption by adding heat exchangers to the originally proposed simple-cycle gas turbines. If they are to be competitive with reciprocating engines, gas turbines must not only be equal or better in performance, but equal or lower in cost. This would require manufacture in large quantities, and it is very likely that the first production models will be in a low-priced high-production automobile.

THE interest of automobile manufacturers in the gas turbine for vehicle propulsion has continued during the past six years. Progress has been steady although not rapid. Many experimental models have been built and indications are that production models may be available within the next decade. The automotive models under development are largely in the 200 to 250-hp size running at 35,000 to 55,000 rpm. Special effort has been devoted in recent years to improving the fuel consumption by adding heat exchangers to the originally proposed simple-cycle gas turbines. Thus a great deal of the effort of the past six years has been devoted to the seal problems involved in designing rotary heat exchangers or the problems of manifolding for stationary heat exchangers.

Probably no group of engineers is working so diligently to eliminate their present assignments as the development engineers and designers of present-day automotive reciprocating engines. With each model of reciprocating engine which has an increased horsepower, a design requiring higher octane fuels, a fuel-induction system which is more costly, or an engine with larger air filters, the ability of a gas turbine to compete economically with the reciprocating engine becomes more evident. A 200-hp gas turbine can have about the same torque characteristics as a 300 to 350-hp reciprocating engine using automatic transmissions. Although it appears to be very difficult to build a 50-hp gas turbine to compete with reciprocating engines, in the 200 to 250-hp size, efficiencies rise in an area where the gas turbine becomes more competitive. Gas turbines can use fuel oil as readily as 100-octane gasoline. The gas turbine can have many fewer parts than the reciprocating engine, making it more competitive on a production-cost basis, especially when reciprocating engines use expensive fuel-injection systems. The bulk of the gas turbine increases considerably when a heat exchanger is added. However, when excessively large air filters are added to reciprocating engines, the gas turbine becomes more competitive in size.

Although it might seem that gas turbines should first appear in sports cars, trucks, and buses, it is more likely for economic reasons that production models of gas turbines will first appear in high-production low-cost automobiles. Intercity trucks and buses operating on superhighways at almost continuous maximum horsepower would seem to be a natural application for the gas turbine. In the case of sports cars, a higher first cost might offer a market. However, if gas turbines are to be competitive with reciprocating engines, they must not only be equal or better in performance, but equal or lower in cost than reciprocating engines. Gas turbines must be manufactured in large quantities in order to get the cost down. Therefore it is very likely that the first production models will be in a low-priced high-production automobile.

Maintenance of an automotive gas turbine will be less than that of present reciprocating engines if the experience with other applications to aircraft, pipeline pumping, production of electrical energy, or marine use is a guide. Turbine wheels could well have a life of 100,000 miles and be replaced at that time with less difficulty than putting new rings into a reciprocating engine. Various manufacturers in both the United States and England have predicted the appearance of production models in anywhere from 2 to 7 years. The technical problems of producing an efficient gas turbine have been solved, but manufacturing problems, the reduction of costs, and the design of a chassis and body specifically for gas turbine use are still to be accomplished.

General Motors[1]

"In keeping with the General Motors policy of initiating research programs for all new types of motive-power plants, a program in small automotive gas-turbine engines has been pursued in earnest for the past decade.

"This program has resulted in the design, fabrication, and development of three small gas-turbine engines and the installation of these engines in vehicles with widely varied characteristics. These range from futuristic "research laboratories on wheels," to very practical production commercial vehicles.

"Whirlfire Turbo-Power GT-300, Turbocruiser Installation. Early work in the field of small gas-turbine engines revealed apparent advantages and disadvantages that could only be evaluated by building and testing an actual engine. The design of the first Whirlfire engine, the GT-300, was based on the general concept that many problems would arise in the development program and that mechanical simplicity and reliability should be emphasized over fuel economy. The GT-300 engine, Fig. 1, was a simple-cycle, nonregenerative design with a free-power turbine. It was rated at 325 hp at a design point of 26,000 rpm on the gasifier section, 12,500 rpm on the power turbine, 3.5:1 pressure ratio, and a 1500-F turbine-inlet temperature. The engine utilized a single-stage centrifugal compressor, single-stage axial turbines, and a single combustor designed to burn kerosene or diesel fuel.

"Although extensive test-cell evaluations were made with the basic engine, an actual vehicular installation was considered desirable for performance, control, and maintenance studies. A GMC transit bus, the Turbocruiser, was selected as a test vehicle because: (*a*) The engine could be installed in the existing

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, May 29, 1958. Paper No. 58—A-46I.

[1] The description of General Motors Gas Turbine Development was written by W. M. Spitzer, GM Research Staff.

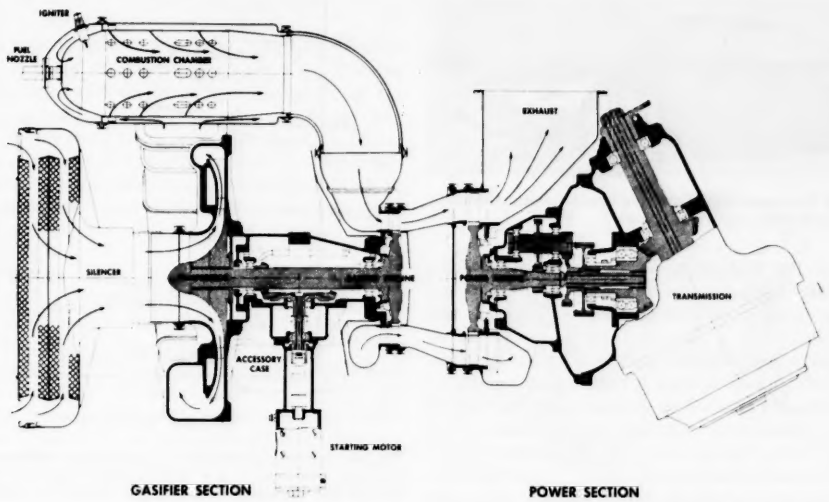

Fig. 1 Schematic arrangement of the GT-300 Whirlfire gas-turbine engine rated at 325 hp and used in the Turbocruiser

compartment with ample space for special test instrumentation; (*b*) the vehicle could be ballasted to represent a wide range of service conditions; (*c*) the seating capacity could be used to advantage in demonstrations; and (*d*) ample space was available to accommodate special test equipment and observers.

"Whirlfire Turbo-Power GT-302, Firebird I Installation. Before the GT-300 engine could be installed and tested in a vehicle, another version was built which was designed to be as compact, lightweight, and efficient a unit as possible without any change in the generally conservative design philosophy. The result was an engine rated at 375 hp at the same design conditions. The GT-302 engine was designed specifically for the Firebird I, a high-performance-vehicle intended for test-track rather than highway use, which allowed great latitude in its design. The power package was mounted in the rear of the vehicle and was composed of the engine, transmission, and rear axle complete in one unit. The transmission utilized planetary gearing and provided two speeds forward and a reverse. No fluid member was used.

"These installations, the Firebird I, the first American gas-turbine automobile, and the Turbocruiser, the world's first gas-turbine-powered bus, demonstrated the feasibility of gas turbines as vehicular power plants and confirmed the claims of paper studies. The desirable features indicated extremely smooth power application, excellent performance, favorable weight-to-power ratio, low oil consumption, an omnivorous fuel characteristic, reliability, and durability. Some of the paper problems that did not materialize, or were proved of minor consequence included engine noise, engine-compartment heat and ventilation requirements, high speed of rotation, and engine controls. After many miles of operation only two disadvantages remained to prevent a favorable over-all performance comparison of the gas turbine with contemporary vehicular power plants: (*a*) High fuel consumption, which was expected when the simple cycle was adopted for the first design; and (*b*) a rather sluggish acceleration characteristic of the engine between idle and 70 per cent gasifier speed.

"Whirlfire Turbo-Power GT-304 Engine. With the background experience of both the mechanical design of a small gas-turbine engine and operation of high performance and commercial types of vehicles, a new engine design was undertaken and slanted toward solving apparent shortcomings. To drastically reduce fuel consumption, efforts were directed toward improving component efficiencies, minimizing flow losses, raising peak operating temperatures, and the incorporation of a regenerator in the cycle. A stationary heat exchanger is a familiar piece of equipment but reasonable effectiveness can only be obtained in a bulky unit. A rotating heat exchanger, on the other hand, presents high effectiveness coupled with relatively small size and weight.

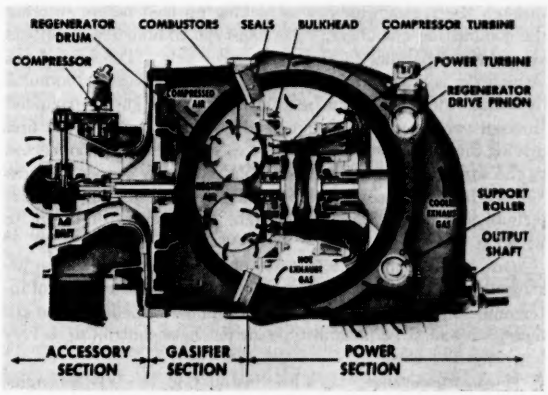

Fig. 2 Final engine arrangement of the GT-304 200-hp regenerative gas-turbine engine used in the Firebird II and Turbo-Titan truck installations

"For a two-shaft gas turbine, the engine-acceleration delay is a direct function of the moment of inertia of the rotating elements of the gasifier section of the engine and an inverse function of the amount of energy in the working fluid available for acceleration. As a result, the acceleration delay could be reduced by lowering the moment of inertia of the rotating elements of the gasifier, especially the turbine wheel and compressor rotor, or increasing the turbine-inlet temperature during acceleration periods, or both. Careful designing effected improvements in both areas, although the ultimate strength and repeated-thermal-shock properties of materials available for turbine disks and buckets places a very practical limitation on allowable temperatures.

"A rotating-matrix regenerator was incorporated into the design of a new engine which was rated at 200 hp at 35,000 rpm on the gasifier, 28,000 rpm on the power turbine, a pressure ratio of 3.5:1, and a turbine-inlet temperature of 1625 F. In designing

Fig. 3 The Firebird II, a 4-passenger futuristic vehicle which utilizes the GT-304 200-hp regenerative gas-turbine engine

the GT-304 engine, Fig. 2, the over-all power package was given prime consideration. Much thought was given to the design and location of each component in order to achieve a compact, well integrated assembly.

"Air enters a radial-flow compressor from the left. After passing through the diffuser, it is diverted at right angles from its radial path and discharged axially into a plenum chamber housing the regenerators, combustion chambers, and turbines. The plenum is divided into a high-pressure and low-pressure section by seals and a center-bulkhead assembly. Two drum-shaped regenerators, rotating about a horizontal axis perpendicular to the main engine-shaft axis, pass from the low-pressure or exhaust side of the plenum to the high-pressure side through close-clearance floating seals. The compressor-discharge air passes radially through the regenerator drums picking up heat before entering the combustion chambers. The can-type combustion chambers are arranged parallel to the regenerator axis. The heated gas leaving the combustors is diverted into the turbine-inlet annulus by symmetrical transition sections. The hot gas is expanded through two mechanically independent turbine stages, the first driving the compressor and accessories, and the second connected to the working load through suitable reduction gears. Gas exhausted from the second turbine is diverted into the low-pressure plenum and passes radially through the regenerator drum, to which it gives up heat.

"One of the important features of this integrated engine arrangement is the direct air-flow path and complete absence of interconnecting ducts. The minimum number of bends in the air-flow path and large plenum-type air passages contribute to very low pressure losses.

"Firebird II Installation. As a first installation, the GT-304 engine was mounted in the Firebird II, a 4-passenger futuristic vehicle shown in Fig. 3. The engine is mounted forward in the conventional location, but the transmission is located behind the passenger compartment in line with the rear wheels. Typical engine performance is shown in Fig. 4.

"One of the most surprising features of the GT-304 engine is the very low noise level. Performance of the GT-304 in the Firebird II showed significant improvement over the Firebird I and Turbocruiser installations. The fuel consumption was reduced by over 50 per cent. The engine-acceleration delay was notably less and low-speed acceleration was enhanced by the use of an automatic four-speed planetary-type transmission.

"Turbo-Titan Installation. With the background experience in gas-turbine-engine potential and the availability of a reliable 200-hp engine, it appeared that the development program had reached a stage where full-scale comparison should be made with existing equipment. For this purpose the GT-304 engine was installed in a production heavy-duty Chevrolet truck-tractor (Model 10413) which incorporated a tandem rear axle, a six-speed automatic transmission, and a three-speed transfer box. The standard engine for this vehicle was a 322-cu-in. overhead valve V-8 rated at 195 gross hp.

"The most important advantage shown in tests with the Turbo-Titan was a significant gain in performance. The gas-turbine

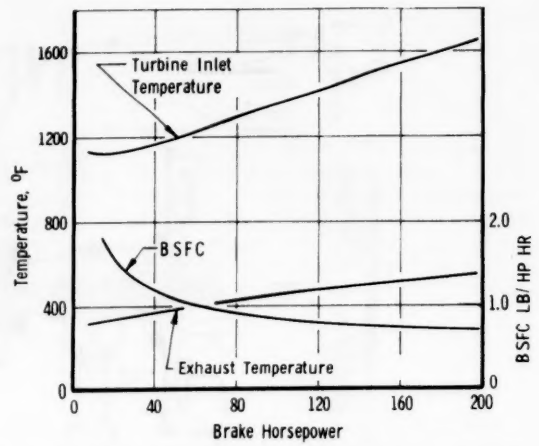

Fig. 4 Typical performance parameters—turbine-inlet temperature, exhaust temperature, and brake-specific-fuel-consumption figures—for the GT-304 regenerative gas-turbine engine, as plotted against brake horsepower

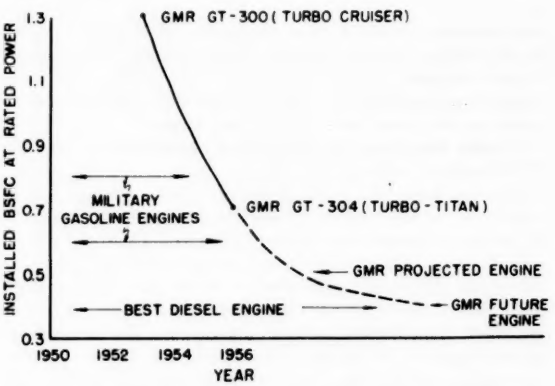

Fig. 5 Past, present, and expected future progress in the efficiency of small gas-turbine engines at General Motors Research Staff

truck demonstrated 27.5 per cent improvement in level-road, full-throttle acceleration to 40 mph. A 17 per cent improvement in gradability was demonstrated. These improvements were due in most part to having more usable horsepower available due to the rising torque characteristics and resulting broad horsepower peak of the free turbine in the gas turbine.

"Over-all fuel-economy characteristics of the Turbo-Titan were still below those obtained with the production truck; however, significant advances had been made. The improving brake specific-fuel-consumption figures for existing, proposed, and projected small gas turbines at General Motors are shown in Fig. 5.

"As new applications of existing procedures, techniques, and cycle arrangements are made, the area in which the small gas turbine is competitive with piston engines will be enlarged significantly without the requirement of a major scientific or engineering break-through. The automotive gas turbine is at the threshold of a rapidly accelerating development cycle and the highest potential of the small gas-turbine engine remains untapped, waiting for future exploitation by clear-thinking, practical, and ingenious scientists, engineers, and technicians."

Chrysler

According to a 1957 progress report on the Chrysler Corporation's development of an automotive gas turbine by Huebner [3],[2] the preliminary studies on which the first model of the Chrysler turbine was based were: Turbine efficiency, 85 per cent; burner efficiency, 95 per cent; compressor efficiency, 80 per cent; cycle pressure loss of 6 per cent; 3 per cent leakage loss indicating rotary heat exchanger; burner turbine-inlet temperature, 1600 F; and a pressure ratio of 4 to 1 for the cycle. The following quotation from the Huebner paper describes the first model of the Chrysler gas turbine.

"Fig. 6 shows the result of our initial efforts. The air intake at the left of the engine is annular, and the air flows uniformly from its circumference to the compressor-inducer which has two rotating axial stages integral with the cast radial compressor. The intake passages are lined with insulating material to prevent heat transfer to the incoming air. The radial diffuser and collector delivers the air at low velocity to the engine-top cover which directs it downward through the regenerator to a plenum chamber communicating with the space surrounding the burner combustion sleeve. The flow after heating is introduced into the burner and directed downward through the burner sleeve and a conductor pipe to a vortex chamber communicating to the first-stage nozzle. After flowing through the two turbine stages the exhaust gas passes upward through the regenerator and is again directed downward, by the top cover, to the exhaust pipe (not shown since it is located diametrically opposite to the burner in this drawing). The housing is cast iron. Both the engine-top cover and the burner cover are cast aluminum. Accessories are driven from a reduction gear mounted at the front of the engine and meshed with a pinion on the gas-generator shaft. The accessory-reduction-gear ratio is 8.3:1. The gear drives a shaft which runs parallel to the gas-generator shaft, driving the fuel pump, regenerator, and electric starter-generator. Final reduction of 12.5:1 is through a 2-step reduction gear at the rear end.

"The regenerator is a circular built-up disk 18 in. in diam and 3 in. thick and is currently operating with a heat-recovery effectiveness of 83 per cent at full load and 87 per cent at quarter load.

"This effectiveness was obtained only through careful research and component development. However, it by no means represents what we consider to be the pinnacle of regenerator development. We have already exceeded this figure by three percentage points and are working with considerable success on even higher heat recovery.

"The compressor used in this engine operates at 4.25:1 pressure ratio, and measured adiabatic efficiencies of 78 per cent have been obtained. Construction of this original compressor-impeller was quite interesting. The impeller itself is an aluminum casting with the vanes fully shrouded. The inducer section consists of two rows of cast-aluminum blades loosely placed in slots in the shaft and retained radially by a steel shroud shrunk to the eye diameter of the impeller shroud. The shaft is steel and is of the built-up type with a calculated natural frequency of 77,000 rpm. A roller bearing in the front end and a ball bearing at the rear are jet-lubricated. Shaft seals may be seen directly behind the impeller and on the hub of the compressor-drive turbine. Operating speed of this unit is in excess of 51,000 rpm and, therefore, components and the entire assembly are balanced to quite close tolerances. The entire assembly is made by stacking the parts from either end and is retained by a through bolt which passes down the center of the shaft, the drive of the assembly being taken by friction between the various parts.

"The rear is assembled similarly to the power turbine and is supported at its forward end, behind the power turbine, by a ball bearing. The rear end of the shaft is supported by the reduction-gear drive pinion which is straddle-mounted in roller bearings.

"The engine as shown in Fig. 6 is the type first operated. It was later shown publicly in March, 1954. With changes not shown in this figure, it is the engine which we used in our transcontinental run, March, 1956."

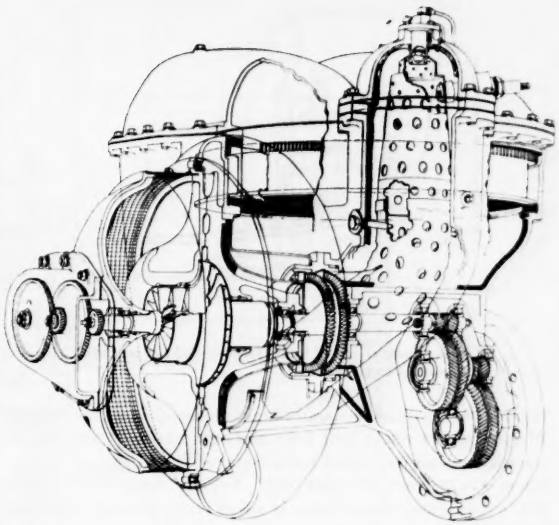

Fig. 6 Schematic diagram of the Chrysler initial gas-turbine design

The transcontinental run was made with the regenerator operating for the entire trip. The over-all economy ran between 13 and 14 miles per gal. This included the frequent extended periods of idling which occurred because people wanted to look at the car. The engine was not shut off from New York to San Bernardino.

Chrysler Corporation[3] has been actively engaged in turbine work for a period of more than ten years. The resulting experience has made it possible to build and operate turbine engines which rival more conventional automotive engines and will soon surpass them. We believe that the gas turbine has the greatest potential in terms of efficiency, simplicity, and lighter weight. It has a torque curve which can match the power requirements of an automotive vehicle much more closely than is possible with a reciprocating engine. In addition to having greater potential thermal efficiency and better output characteristics for an automobile, the turbine is a durable and reliable engine. It is much lighter and has 80 per cent fewer moving parts than a reciprocating engine, making it easier to build and maintain once the development stage is passed.

A turbine ride is essentially vibration-free because combustion is continuous and because the few moving parts rotate smoothly on ball or roller bearings. Being air-cooled, the turbine requires no radiator or liquid-cooling system. The closest thing to a radiator on the Chrysler turbine engine is a small cooling tube for the lubricating oil. The electrical system consists of a storage battery, starter generator, coil, breaker, and a single spark plug which is needed only in starting. The transmission component in the present test model is used for the sole purpose of providing a reverse gear.

Fuel consumption has been reduced to equal that of piston engines, and there are no octane requirements. Properly developed, it will operate on almost any fuel that will flow through a pipe. The principal reason for the Chrysler gas turbine's modest appe-

[2] Numbers in brackets designate References at end of paper.

[3] The following portion was written by J. P. Duchamp, Engineering Division of Chrysler Corporation, April, 1958.

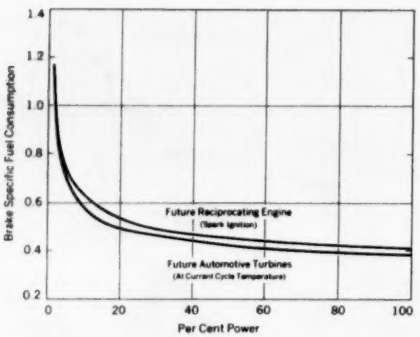

Fig. 7 Comparison of future turbine and reciprocating engine fuel consumption

tite for fuel is the regenerator unit mounted atop the turbine which recovers waste heat from the exhaust gas to preheat incoming fresh air on its way to the combustion chamber. Its effectiveness is in excess of 90 per cent. Not only does fuel go further but exhaust-gas temperature is below that of a conventional passenger car.

The practicality of the Chrysler turbine engine was best demonstrated in March, 1956, when a standard Plymouth automobile powered by one was driven from New York to Los Angeles. That turbine unit, together with its set of reduction gears, was only 32 in. long, 33 in. wide, and 28 in. high. It was based on our original design which has since been superseded.

The materials used in our early engines were the best that could be obtained. Our progress to date on the utilization of readily available noncritical elements has indicated that current operating temperatures can be maintained for satisfactory operating life without the use of cobalt; the required nickel content would likewise be well within a practical range from the standpoint of availability.

The turbine facts already described should not be new to the industry. They were presented by Huebner at an SAE Meeting in 1956 [3]. Several predictions were then made with regard to the performance of Chrysler gas turbines and the likelihood of specific improvements. These have been demonstrated to be sound. The efficiency of components has been increased, fuel consumption reduced, the problems of acceleration lag and engine braking overcome, and new materials developed. In spite of the optimism represented in Huebner's curves for specific fuel consumption, Fig. 7, steady progress has been made in the direction predicted. While mass production of turbine engines for passenger cars has not been achieved, practical results are increasing.

When gas turbines will freely compete with piston engines in automobiles depends on the solution of various engineering, manufacturing, and supply problems. However, gas turbines will someday be produced commercially at costs comparable to those of today's conventional piston engines. Our confidence is based on the progress already achieved in establishing the turbine as a practical automotive-power source.

Ford[4]

"The Ford automotive-gas-turbine development program was started in 1952.

"From the beginning, attention has been focused on the conventional automotive-turbine cycle. A compressor pressure ratio of approximately 4.0 is used and a turbine temperature of approximately 1500 F.

"Component-development work included all of the major elements with particular attention to the compressor and the rotary heat exchanger. A compressor efficiency of about 80 per cent was attained at a pressure ratio of about 4.0. Heat exchanger effectiveness of approximately 80 per cent was attained.

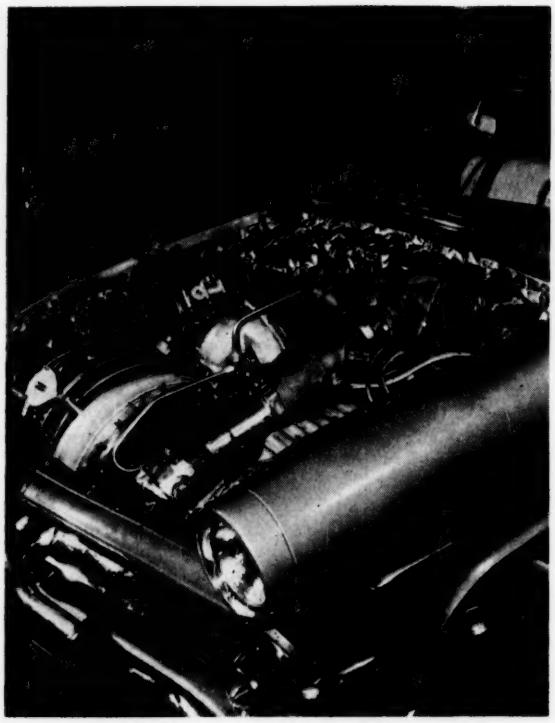

Fig. 8 Ford automotive gas turbine

Fig. 9 Ford truck gas turbine

"In 1954, the first engine was built and installed in an otherwise standard 4-door Ford sedan, Fig. 8. This unit was purely experimental and was assembled from components which were available from the component-development program. After a brief development program on the test track, it was retired.

"Application studies indicated that heavy-duty trucking service would be more suitable as an initial application of the turbine engine. Accordingly, a revised version was designed in 1955 and dynamometer tested in 1956. Late in 1956 this engine was installed in a TC-800 Ford truck, Fig. 9. This truck underwent extensive road testing and development during 1957 and early 1958. While no performance figures have been released, the gradability exceeded that of a comparable conventional-engined vehicle.

"The truck engine is designed for 160 hp net. The pressure ratio

[4] The following portion was written by Kenneth Bodger of the Ford Motor Company. Mem. ASME.

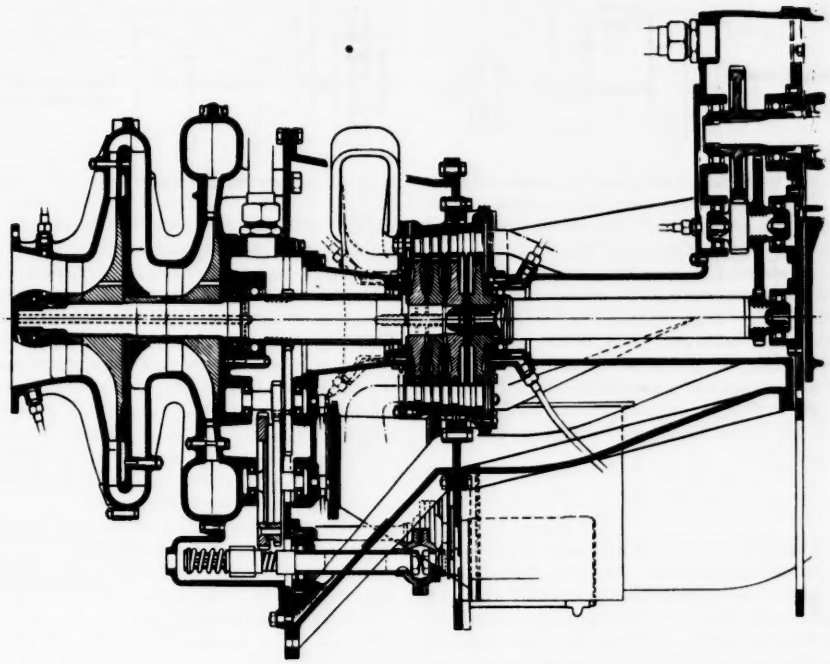

Fig. 10 Austin automotive gas turbine

is 4.0, and the turbine-inlet temperature about 1500 F, as in the previous model.

"More recent work is being carried out with the objective of both truck and passenger-car application in the development program. As of the publication date, however, no details are available for publication."

According to a paper published by Beaufrere [4], the first experimental gas turbine built by Ford used a turbine temperature of 1600 F, a pressure ratio of 4:1, a rotary heat exchanger, a radial-flow turbine with variable-area nozzles, a turbine efficiency of better than 80 per cent, and a compressor efficiency of 79 per cent.

Austin

The Austin Motor Company of England began an experimental program for the development of an automotive gas turbine in 1949 [6].

Although the pressure ratio is 4:1, they used a two-stage compressor to reduce rotational speeds and stresses, thereby enabling the use of higher cycle temperatures or lower-cost alloys in the compressor. The unit was designed for 120 hp with an air flow of 3 lb per sec. A stationary type of heat exchanger was selected and thereby the effectiveness was limited to 80 per cent for a counterflow or 65 per cent for a crossflow heat exchanger. Cycle pressure drops of 10 per cent were allowed for and target efficiencies on the compressor were 80 per cent, and on the turbine, 85 per cent. A cycle temperature of 1475 F was used. The general arrangement of the units is shown in Fig. 10. Three axial stages of turbine are used to drive the two centrifugal stages of the compressor. A single-axial-stage work turbine is directly connected to the reduction gear. Fig. 11 shows specific power and specific fuel consumption for actual test conditions and for target conditions using thermal ratios of 0.65 and 0.80. To achieve competitive fuel consumption with present-day reciprocating engines, a higher thermal ratio and more efficient compressors will

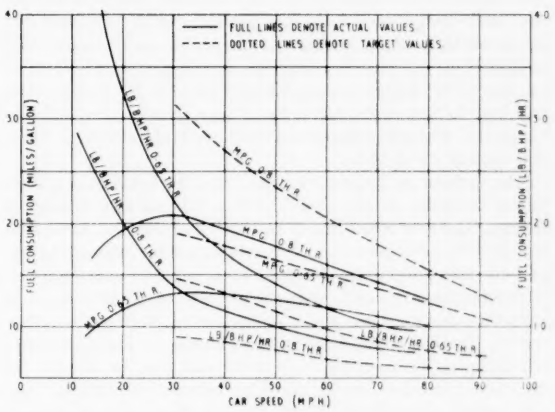

Fig. 11 Road-load fuel consumptions—actual and target—Austin automotive gas turbine

be required. Sir Leonard Lord of the Austin Motor Company has been quoted as reporting that a 250-hp Austin turbine will be in production in 1959.

Rover

The Rover Company, Ltd., of England was the first automobile manufacturer to operate a car with a gas-turbine drive—in 1950. In June, 1952, the Rover gas-turbine car established a speed record of 152 mph. The latest model for which data are available has a 110-hp turbine operating at 52,000 rpm with a pressure ratio of 3.85, a gas temperature of 1525 F, and an air flow of 2 lb per sec. It drives a car through a 4-wheel drive and has the engine in the rear. It uses a single-stage 17-vane centrifugal compressor with a single-stage axial turbine to drive the compressor and a single-stage turbine operating through a 7.45 gear ratio to

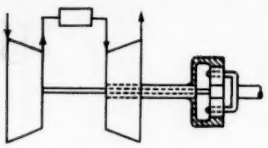

Fig. 12 Differential gas turbine

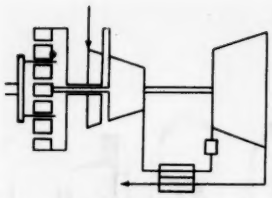

Fig. 13 Split-compressor planetary-drive automotive gas turbine

drive the vehicle. It is capable of attaining 60 mph in 10.5 sec and 80 mph in 17.7 sec. The latest model incorporated a plate-type heat exchanger using counterflow and mounted on top of the engine. Fuel-consumption figures for this turbine are not competitive with present-day reciprocating engines of English manufacture. Fuel consumptions are 12.4 miles per standard gal at 40 mph, 12.9 miles per gal at 60 mph, and 11.5 miles per gal at 80 mph in a lightweight English car. S. B. Wilkes, chairman of the Rover Company, has been quoted as saying: "Work on the development and improvement of a gas-turbine car, exhibited at the Earls Court show in 1956, is continuing satisfactorily but it is still too early to decide on our future policy regarding production of a car of this type."

Fiat

The Corse G. Angnelli of Torino, Italy, began a gas-turbine development for automobiles in 1948. Data on their development was published in a paper presented at the ASME Gas Turbine Power Conference in 1957 [7]. This unit was designed for 200 hp without a heat exchanger. It is intended to be mounted in the rear of a vehicle and has a two-stage centrifugal compressor with a 7:1 pressure ratio. The turbine temperature is 1475 F. One feature of this engine is the use of separate shafts for the turbine and compressor of the gas generator with two axial-flow stages on the turbine driving the compressor and a single-stage axial-flow power turbine connected to the gear box. It incorporates three combustion chambers. Because the turbine does not have a heat exchanger it has a very poor specific fuel consumption. Published figures are 0.9 lb per hp-hr at 200 hp, 1.3 at 100 hp, and 1.8 at 50 hp.

The Caterpillar Tractor Company and the International Harvester Company of the United States, Socema and Renault of France, Saurer of Switzerland, and Volvo of Sweden, have been reported to have gas-turbine projects under development but as of 1958 have not published any progress or performance reports.

Several recent papers suggest the use of a differential or planetary gear connecting the compressor and turbine for vehicle drives. In 1955, Hutchinson [8] showed the advantages of using a differential between the compressor and turbine, Fig.12. As the output speed of the differential is reduced, the turbine speed will reduce more gradually than the turbine of a single-shaft machine, the output turbine of a two-turbine split-shaft power plant, decreasing to about 50 per cent speed when the output speed is zero. At one half its maximum speed the turbine is still operating in its efficient range and is capable of providing 75 per cent of its maximum power at 150 per cent of its maximum torque. Reducing the output speed to zero provides a 50 per cent increase in the power available to drive the compressor which would result in a rapid acceleration of the compressor shaft. This arrangement improves fuel consumption at part load and reduces the time for acceleration of the compressor turbine. However, the torque multiplication is not as high as that for a split-shaft turbine. Martinuzzi [9], however, pointed out that by overspeeding the compressor, increased torque multiplication can be had at zero output of speed. In 1958, Bowden and Hryniszak [10] suggested the use of a split compressor connected through a planetary gear rather than a split turbine as illustrated in Fig. 13. At full output speed and full power the first-stage compressor is at zero or low speed. The cycle pressure ratio is low but with the addition of a regenerator the fuel consumption is low. At low output speeds, the first-stage compressor increases its speed adding pressure ratio to the cycle and increasing the torque multiplication of the turbine. Under these conditions the turbine is operating at optimum efficiency and good fuel consumption. Thus with this arrangement one gets high torque multiplication at stall with a fairly flat fuel-consumption curve. To date no company has indicated the construction of a gas turbine using planetary gear connecting components, probably because of the complicated nature of planetary gears. The high-speed turbine shaft can be connected directly to the sun gear and the low-speed output shaft connected directly to the planet carrier thus alleviating high speeds in the planets.

The competitive position of the gas turbine for automobiles depends largely upon the construction of a heat exchanger, if vehicles running much of the time at part load are to have good fuel consumption. This is most easily obtained by adding a heat exchanger of high effectiveness, 80 per cent or better. Rotary heat exchangers are capable of high effectiveness in relatively small sizes. This is accomplished by using passages of very low hydraulic radius, thereby increasing the film coefficients and also increasing the surface area exposed to gases. To keep the pressure drop to acceptable limits the length of matrix passages must be kept short. The major development problem for rotary heat exchangers has been to develop seals which will have low leakage losses and a long life. Stationary heat exchangers of similar size and the same effectiveness can be manufactured if the same hydraulic radius is used. Such small passages, however, make the header through which gases enter and leave difficult to manufacture so that one is confronted with developing a header or a rotary seal. Because the gases pass through the passages of a rotary heat exchanger, alternately throughout the revolution of the exchanger, the exchanger is more or less self-cleaning. In the stationary heat exchanger, gases pass in steady flow in a given direction and when small passages are used there has been much fear that dirt will clog the passages. There has not yet been enough experience on this type of heat exchange to prove or disprove the ability of this type of heat exchanger to remain clean. Progress in both rotary seals and manifolds is being made. One organization reports that seals have been developed which will have a longer life than other parts of the gas turbine. Bowden and Hryniszak [10] show a construction for small-hydraulic-radius stationary heat exchangers made entirely of stampings and furnace brazed.

Von der Nuell [11] discusses design improvements for small gas turbines, and, although the paper does not discuss vehicle gas turbines, many of the suggestions and experience gained in the development of small gas turbines can well be applied to the design of gas turbines for automobiles.

If automobile-company management chooses to produce a gas-turbine-powered automobile, a production-model car designed especially for a gas turbine could be produced by 1965 and is likely to have a 200-hp gas turbine of 4:1 pressure ratio, a maximum inlet gas temperature of 1600 F, a single-stage centrifugal compressor of better than 80 per cent efficiency, a rotary or stationary heat exchanger of 85 to 90 per cent effectiveness, a two-stage turbine of 85 per cent efficiency or better, one stage driving the compressor, and one stage driving the output shaft.

Compared with present-day reciprocating engines, size will be no larger, maintenance will be less, fuel consumption will be as good or better, antifreeze will not be required, torque characteristics will be better, less lubricating oil will be used, and kerosene or fuel oil will be burned.

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RAILROAD

Progress during 1957 on the coal-burning gas-turbine power plant being perfected by the Locomotive Development Committee of Bituminous Coal Research, Inc., is reported. Operating experience with the Union Pacific Railroad's 4500-hp General Electric gas-turbine-electric units, and the characteristics of the 8500-hp units on order from General Electric are given. The U. S. Army's 30-ton unit, Electro-Motive Division's free-piston engine, several British Railways developments, Soviet designs, Swiss, Czechoslovakian, Swedish, and French progress are reported.

Coal-Burning Gas-Turbine Power Plant

THE 1957 test program subjected the gas-turbine power plant being perfected by the Locomotive Development Committee of Bituminous Coal Research, Inc., to 1103 hr of operation at load factors far in excess of normal locomotive-operating practice. The general operation of the plant was more satisfactory than in any previous testing program.

With the completion of the 1957 test runs the Locomotive Development Committee has operated 3760 hr of coal-fired time on the locomotive-size Allis-Chalmers gas-turbine unit and an additional 1250 hr of coal time were run on a Houdry unit bringing the total coal-burning time to 5010 hr.

The Operations Committee of the Locomotive Development Committee recommended the continuance of testing operations at Dunkirk with the following work to be done: (*a*) Test and improve an aerated coal system to use "specification" coal, Fig. 1; (*b*) make improvements to coal pump, combustors, and fly-ash separators; (*c*) alter and reblade the LDC-Allis turbine as necessary, Fig. 2; (*d*) make running tests with rebladed turbine and improved components, Fig. 3.

The Locomotive Development Committee approved these plans and made funds available for commencing this work in January, 1956.

Tests started on January 9, 1957, and 310 hr were run by February 28 at which time operations were stopped as the result of a strike in the Alco Products Plant at Dunkirk, N. Y., where the test unit is located. Work was resumed on May 27 and in the next 51 operating days ending August 14, 793 hr were run making a total of 1103 hr.

The pertinent details of this operation are summarized as follows:

Summary of operations—1957 test

Item	Pittsburgh No. 8 coal	Pocahontas No. 3 coal	Accumulative results
Hours run	753:11	349:39	1102:50
Tons burned	1143	520	1663
Average coal rate, lb per hr	3030	2975	3002
Hp-hr generated	2,224,149	1,041,957	3,266,106
Lb coal per hp-hr	1.03	1.00	1.02
Average load, hp	2953	2980	2960
Load factor, per cent	83.42	84.18	83.66
Thermal efficiency, per cent, based on low-heat value	19.78	18.34	19.32

Operation of Coal System. During the 1100-hr test the coal system, Fig. 4, including the aerated feed, coal pump, and pressurized pulverizer performed in a very satisfactory manner.

The aerated-coal system handled 1663 tons of coal during the test period and demonstrated that if fed "specification" coal it could adequately sustain operations. During the test period the coals used and their "as-received" size were as follows:

Sieve size	Pittsburgh No. 8, per cent by weight	Pocahontas No. 3, per cent by weight
+28	49.2	54.1
−28 +60	27.5	20.9
−60 +100	9.6	7.4
−100 +200	8.2	8.7
−200	5.5	8.9
Surface moisture	3.8	...
Total moisture	...	3.5

Coal Pumps. During 1956 a new design of coal pump, designated CP5A, was made incorporating changes necessary to eliminate weaknesses exposed in previous tests.

Two new CP5A coal pumps were manufactured using improved materials. The interior and all parts exposed to air-coal mixtures were chrome plated.

One of these pumps was installed prior to the start of the 1100-hr test and ran the entire test program without any failure or maintenance.

LDC-Riley Pressurized Pulverizer. As a joint project in 1955, LDC and Riley Stoker Company developed a pressurized mechanical pulverizer of the attrition type. One pulverizer rated at 3000-lb capacity with 57 Hardgrove-grindability coal was built and the Locomotive Development Committee undertook the testing and final development. Considerable testing on this pulverizer was done during 1956 and Riley co-operated in making many changes designed to improve particle fineness and reduce horsepower input.

This pulverizer was used during the entire 1100-hr test period and averaged an output of 3000 lb per hr over the entire test

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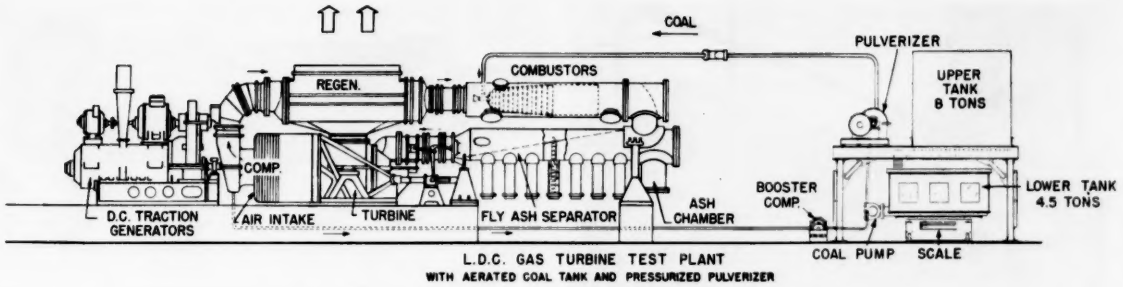
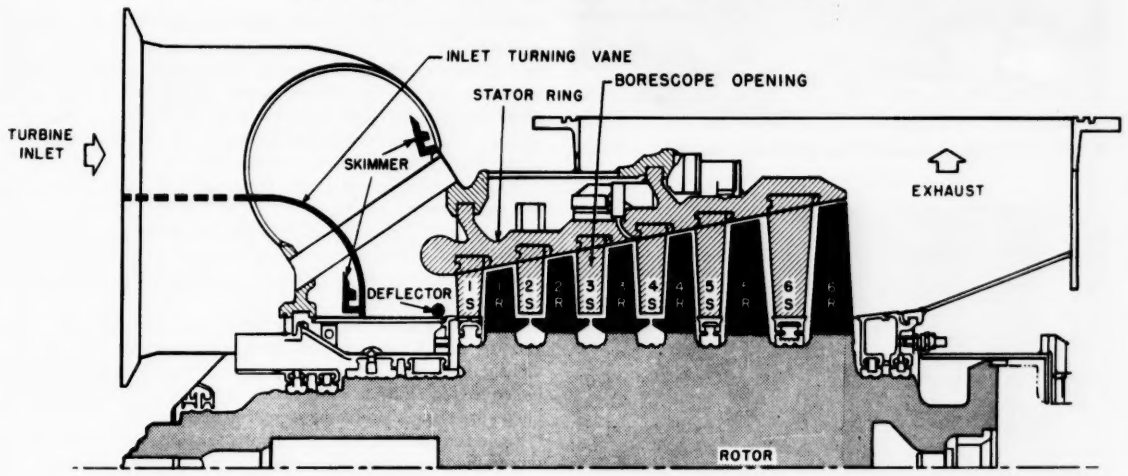


Fig. 1 Diagram of LDC 1957 locomotive coal system and power plant



NEW ROTOR BLADES AT LOCATIONS R1, R2, R3 AND R4. ORIGINAL BLADES AT LOCATIONS R5 AND R6

INNER STAGE SHROUD AND SEAL RINGS REMOVED AT LOCATIONS S2, S3 AND S4. NO NEW STATOR BLADES ADDED.

DEFLECTOR RING INSTALLED IN FRONT OF FIRST ROW STATOR BLADES TO AVOID CONCENTRATION OF DUST AT INNER ROOT.

SKIMMERS INSTALLED ON THE CONCAVE FACES OF BOTH TURBINE INLETS TO COLLECT AND BLOWDOWN NATURAL DUST CONCENTRATION

Fig. 2 Diagram of blade arrangement for LDC 1957 runs

period, at times as much as 3400 lb of coal were passed through the machine with acceptable fineness.

The pulverizing performance on the test coals was as follows:

Sieve size	Pittsburgh No. 8, per cent by weight	Pocahontas No. 3, per cent by weight
+ 60	0.2	0.04
− 60 +100	0.8	0.3
−100 +200	9.4	3.5
−200	89.6	96.16

Operation of Combustors and Fly-Ash Separator. Combustors. During 1956 the existing two combustors, Fig. 5, were modified to separate the combustion and mixing zones in an effort to minimize distortion. The present two combustors operated throughout the entire 1100-hr test and now have a total of 3630 hr of coal-fired operation plus several hundred hours of oil-fired time. The combustors continued in serviceable condition at the conclusion of the test program. Some warpage takes place in the burning zone and the mixing zone but does not appear to affect combustion efficiency. Combustion efficiency was consistently over 95 per cent with each of the two coals burned in 1957.

From the standpoint of "hours of service" and "combustion efficiency" it would appear that the present LDC combustor design is satisfactory for locomotive use.

Fly-Ash-Separator Performance. During 1956 work was done to improve the efficiency of the Dunlab tube, Fig. 6, and to facilitate the blowdown of ash from the tube annulus. This test work proved that the ash-removal system could be greatly simplified and a manifold system was designed and built as the result of these experiments. An improved method for checking the performance of the blowdown system was designed and built, and functioned throughout the 1100-hr test.

Turbine Operation. During the 1103 hr of the test the turbine operated in a completely satisfactory manner. No turbine maintenance was necessary, no electrical trouble was experienced, and no instrumentation or recording-device troubles were encountered. The ignition system functioned without trouble.

Blade Conditions. It is the feeling of the staff that the excellent performance of the pulverizer in supplying fine coal for combustion coupled with continued high combustion efficiencies, proper functioning of the separator, and improved blade materials has resulted in a remarkable decrease in blade wear. Micrometer readings taken on the rotor blades indicate that the wear rate varies between 0.003 and 0.009 in.

When the gas turbine was opened for inspection it was found that the scalloping and abnormal wear that had previously oc-

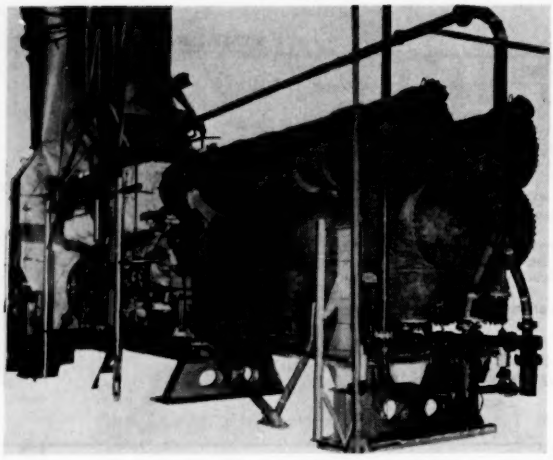


Fig. 3 General view of LDC test plant

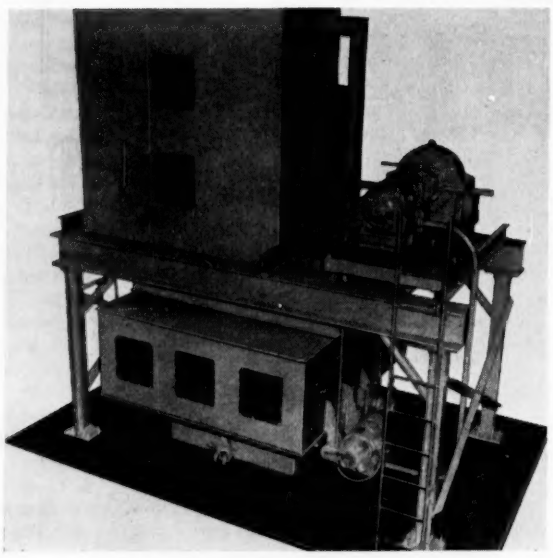


Fig. 4 Photo of LDC aerated-coal system

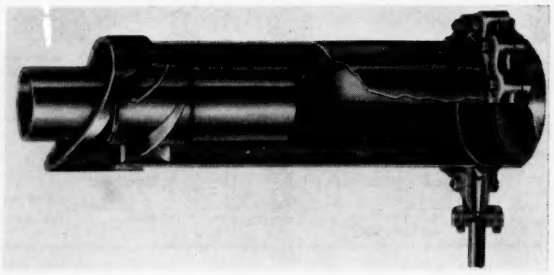


Fig. 6 LDC mark III D Dunlab separator tube

curred on the trailing edges of the rotor blades were not present. An undercutting of the leading edge of the second, third, and fourth-row rotor blades was noted. This is apparently caused by leakage under the tips of the stator blading which, due to its direction, results in high impact velocity at the root of the rotor blades. The total erosion that has taken place in this test is only a fraction of that previously experienced.

When the turbine was opened the rotor blades were found to be free from heavy deposits.

In the 1957 turbine alterations the gland-seal ring was not removed from the number 1 stator row and it is noted that no undercutting occurred on either the leading or the trailing edges of rotor row 1.

Prior to the 1957 test no new blades were added to the stator and the shroud rings were removed from rows 2, 3, and 4 as previously described. The first four rows of stator blades have been in service for 2825 hr. The fifth and sixth-row rotor and stator blades are original and have seen 3760 hr of service.

Since the start of operations with the Allis-Chalmers turbine power plant in 1951, three major test periods have been run. Each test period has resulted in improved blade life.

Economics. The economics continue favorable to the coal-fired gas turbine. ICC Statement M230 shows the 1957 averages for diesel fuel in the Eastern District (including Pocahontas Region) costing 11.42¢ per gal, up 0.38¢ from 11.04¢ in 1956 or a rise of $3^1/_2$ per cent. Oil at 11.42¢ per gal is equivalent to 88¢ per million Btu or coal at $22 per ton. Based on the relative efficiency of diesel (30 per cent) versus coal-fired gas turbine (18 per cent), the break-even point would come at $13 per ton for locomotive fuel.

Current Status. At a meeting of the Locomotive Development Committee in November, 1957, it was concluded that the current business recession would not permit the Locomotive Develop-

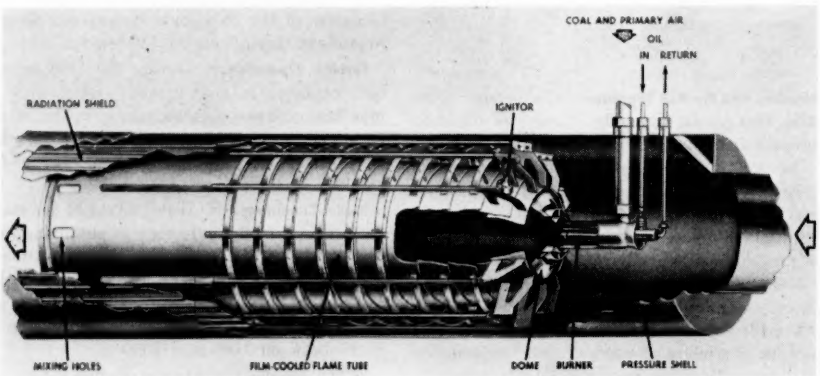


Fig. 5 Cutaway view of LDC combustor

ment Committee subscribers to appropriate the funds necessary to implement the recommendations of the Operations Committee and the Mechanical Advisory Committee.

Sufficient funds have been allocated to place the operation on a "stand-by" basis wherein the key personnel are retained and the test unit "white leaded" until further funds are available.

Gas-Turbine-Electric Locomotives

The 4500-hp Units. During the year 1957, the 25 4500-hp oil-fired single-unit gas-turbine-electric locomotives which were delivered to the Union Pacific Railroad from the General Electric Company during 1952, 1953, and 1954, continued to operate in through-freight service between Ogden, Utah, and Council Bluffs, Iowa, a distance of approximately 1000 miles. Most of the locomotive runs were approximately 500 miles, since locomotives are usually changed at Cheyenne, Wyo., which is approximately midway between Ogden and Council Bluffs.

Mileage performance of these locomotives during the year 1957, compared with that of 1956, is as follows:

	1957	1956	Per cent decrease
Total cumulative locomotive miles since purchase	11,519,019	8,856,803	...
Total locomotive miles in freight service	2,654,863	3,028,147	12.33
Locomotive miles in freight service, average per locomotive per month	8,850	10,094	12.33
Total freight train miles	2,574,225	2,942,047	12.50
Total freight gross ton-miles	9,711,475,000	10,609,216,000	8.46
Percentage of total freight gross ton-miles U.P. System which were hauled by gas-turbine-electric locomotives	11.44	12.14	5.77

By the end of 1957, locomotive 51, which was the first GTE locomotive placed in service, had accumulated a total locomotive mileage of 629,998 since entering service January 31, 1952.

There are two spare gas turbines for these 25 locomotives, making a total of 27 gas turbines. The maximum number of fired hours accumulated in one year by any one gas turbine since 1952 was 5673. The average fired hours per gas turbine for 1957 was 4490.

Since 1952, one of these turbines, Serial No. 95008, has accumulated a total of 24,340 fired hours, 4981 of which were in 1957. This is the highest total hours for any turbine in the group. The history of this gas turbine is interesting and is summarized as follows:

Date	Total fired hours	Remarks
4- 8-52	0	Originally applied to locomotive 52
6- 9-52	983	Removed from locomotive for modification
6-25-52	983	Installed in locomotive 51
7-31-52	1431	Removed from locomotive account of broken thrust link
8-27-52	1431	Installed in locomotive 56
10-31-52	2283	Removed from locomotive account of broken 2nd-stage turbine bucket
12- 5-52	2283	Installed in locomotive 54 after new 2nd-stage turbine wheel and buckets applied
4-10-53	4029	Removed from locomotive account of 2nd-stage buckets seized. Broken metal pieces wedged between buckets and shroud
4-25-53	4029	Installed in locomotive 52
6-10-53	4716	Removed from locomotive 52 for turbine-wheel modification
6-12-53	4716	Installed in locomotive 53
6-18-53	4772	Removed from locomotive 53 account of scored bearings due to lube-oil failure
7- 4-53	4772	Installed in locomotive 53
11-18-53	6584	Removed from locomotive 53 account of excessive vibration
12- 6-53	6584	Installed in locomotive 53
1-20-54	6936	Removed from locomotive 53 account of excessive vibration
3- 5-54	6936	Installed in locomotive 55
3-18-54	6979	Removed from locomotive 55 account of No. 2 thrust-bearing failure
4-17-54	6979	Installed in locomotive 59
3- 1-55	11179	Removed from locomotive 59 account of battery failure which made the DC lube pump inoperative and resulted in wiped bearings
3-15-55	11179	Installed in locomotive 57
4-23-56	16359	Removed from locomotive 57 account of low horsepower
6-10-56	16359	Installed in locomotive 54
3-16-57	20429	Removed from locomotive 54 account of burned fish tails, burned 1st and 2nd-stage nozzles and worn compressor blading
4- 3-57	20429	Installed in locomotive 55
12-31-57	24340	In service in locomotive 55

At each of these removals the turbine was carefully inspected and all parts in need of renewal were replaced.

Review of this history of the locomotive gas turbine which has the most fired hours of any locomotive gas turbine in the world shows a significant trend of longer periods between removals. The maintenance problems are evident, but so is the progress. The life of turbine components has steadily increased. Turbine 95008 which has accumulated the most fired hours, was removed twelve times, more than any other gas turbine. Six gas turbines have been removed only twice for heavy repairs. Since the first GTE locomotive went into service in 1952 there has been a total of 128 gas-turbine removals for repairs up to December 31, 1957. During this time the 27 gas turbines have operated a total of 488,649 fired hours.

At the end of 1956, the average turbine hours per removal for the previous five years was 2969. At the end of 1957, the average turbine hours per removal for the previous six years was 3386. Average number of months between removals increased from 8.4 to 9.4.

As stated in the 1956 report, all GTE locomotives are equipped with insulated fuel tenders. Total fuel capacity of locomotive and tender is 28,410 gal. Operation with fuel tenders has proven satisfactory.

During the year 1957 no significant modifications were initiated. A number of modifications started during the previous year had progressed toward completion on all locomotives.

The 8500-hp Units. The first 15 of the 8500-hp oil-burning 2-unit gas-turbine-electric locomotives were ordered from the General Electric Company by the Union Pacific Railroad in November, 1955. Delivery, which was expected to start in July, 1957, has been postponed because of unexpected difficulties which developed during factory testing of the new design, two-bearing, single-shaft, gas turbines. These difficulties should soon be resolved and the locomotives placed in service during the second quarter of 1958.

As previously reported, these locomotives consist of two units plus a fuel tender. The "B" unit is not capable of operating unless coupled to the "A" unit. The forward or A unit contains the enginemen's cab and controls together with practically all

auxiliary equipment. Power for auxiliary needs is furnished by a 6-cyl 850-hp Cooper Bessemer Model FWB-6 diesel engine. Off the front end of the auxiliary diesel engine is driven a DC generator, one Gardner-Denver Model WBA 9500 water-cooled air compressor, one traction-motor blower which supplies cooling air to all six traction motors on the A unit, and a small battery-charging generator. Off the back end of the auxiliary engine is driven a single radiator-cooling fan, a second Model WBA 9500 air compressor, and a water pump. Two of the three dynamic-brake assemblies are carried on the A unit, a 2500-gal supply of diesel fuel for the auxiliary engine and for turbine starting, air-brake equipment, electric lockers, and battery boxes.

The second or B unit carries the oil-fired, simple-cycle, two-bearing, single-shaft gas turbine, four main-traction generators, generator blowers, reduction-gear assembly, one dynamic-brake assembly, one traction-motor blower supplying cooling air to all traction motors on the B unit, and electric lockers.

The gas turbine has a 16-stage axial-flow compressor, 10 combustion chambers, and a 2-stage turbine. Rated speed is 4860 rpm. Idling speed is 80 per cent of full speed or 3888 rpm. Normal air requirement is 150,000 cfm, or 570,000 lb per hr. The gas turbine will operate continuously, and will have a full-load residual-fuel rate of approximately 750 to 800 gal per hr. Idle fuel rate will be approximately 300 gal per hr. Based on present operation of 4500-hp gas-turbine-electric locomotives, it is expected that the 8500-hp locomotives will require 600 to 700 gal of residual fuel per hr. It is important to operate a gas turbine with as high a load factor as possible due to the increase in thermal efficiency which occurs with lighter loads. This gas turbine is rated at 8500 hp for 6000-ft elevation and 90-F ambient, which corresponds to 10,700 hp for 1000-ft elevation and 80-F ambient. Maximum input which the electrical equipment will accept is slightly over 8500 hp. With 8500-hp input to the main generators the rail horsepower will be 7000.

Air for the 16-stage axial compressor enters the air-intake silencer located on the roof. This air is compressed to 95 psi and has a temperature of approximately 525 F. After passing through the ten combustion chambers, it enters the 1st-stage nozzles at 1450 F. After passing through the 2nd-stage turbine, it leaves the turbine-exhaust duct at a temperature of 825 F and a velocity of 250 fps. The turbine-exhaust duct has an angle of 30 deg with the horizontal. This is expected to be better in tunnels than exhaust ducts on present gas-turbine locomotives which discharge at approximately 45 deg. A thermal efficiency of 20 per cent based on the lower heating value of the fuel is expected at full load.

Each combustion chamber consists of a cap, one piece louver-type cylindrical liner, and a "fish tail" or transition piece. Combustion chambers are folded back over the axial compressor.

A number of shaft-driven accessories are mounted on the turbine reduction gear. These include atomizing-air compressor, lube pump, fuel regulator, tachometer, fuel pump, and fuel-flow divider-control pump. An auxiliary lube-oil pump and a vapor extractor are motor-driven.

The turbine casing is split along the horizontal center line so that the top half can be removed for servicing. Since the turbine, gear reduction, and generator assembly has a width of 10 ft, there is no aisleway. Suitable ladders and roof walkways are provided. This section of car body over the turbine assembly is removable. Five sliding side-inspection doors are provided on each side for use of maintenance forces at terminals.

Turbine starting is accomplished by a 20-step air-driven controller which substitutes mechanical interlocking in the form of cams for electric interlocking. This device simplifies control functions and should eliminate considerable electrical trouble and maintenance. Turbine starting is accomplished by pressing a button on the engineman's control panel. Once this button is depressed, the turbine starting sequence automatically follows through the complete cycle. Approximately 9 min are required before the locomotive can be moved with turbine power. The minimum sustaining speed of the turbine is 55 per cent or 2680 rpm. Turbine cranking requires 400 hp from the diesel-driven generator.

A similar procedure is followed to shut the turbine down. The shutdown sequence includes a cool-down period since experience has shown the necessity of uniform cooling of the turbine-shaft assembly. The locomotive can be hostled, or the turbine cranked at any time during the cool-down period.

The diesel-engine-driven DC generator furnishes power for hostling, turbine cranking, traction-motor field excitation during dynamic braking, and emergency heat for the fuel tender. When hostling, only two traction motors are used, giving a speed of up to 20 mph on level track.

Both the turbine lube-oil system and the diesel-engine cooling-water system are cooled by a single fan driven from the diesel-engine shaft by means of an eddy-current clutch and right-angled gear box.

Either of the brake-air compressors can be disengaged from the diesel-engine drive shaft by means of special flexible couplings.

The excitation-control involves a unique system of printed-circuit cards with encapsulated static components. Each card is keyed to avoid improper placement. This system has been designed to facilitate trouble shooting and maintenance.

The A and B units are connected with 20 power-cable jumpers, 4 multiple-unit-control jumpers, a diesel-fuel supply line for turbine starting, two coolant-water hoses, air-brake hose lines, and sander-actuating hose lines.

The complete gas-turbine power plant, gear-reduction box, and generator assembly is 41 ft $7^{7}/_{8}$ in. long, and weighs 155,000 lb. This complete assembly is supported on four trunnions which provide for $1^{1}/_{4}$-in. expansion. A single buff pin projects from the turbine-assembly bed plate and engages a hole in the locomotive underframe. This pin transmits all buffing loads to the underframe.

Four main-traction generators, mounted back to back in pairs, are bolted to a platform integral with the gear case and driven at 1050 rpm through reduction gearing from the compressor-inlet end of the gas turbine. Four exciters are gear-driven off the ends of these generators. Also gear-driven off the main generators are pulleys which belt drive the single traction motor which supplies cooling air to the six traction motors on the B unit, and the two generator blowers. The generator blowers supply clean air to the center of each pair of generators where it divides and passes over the rear-end windings first, then over field-excitation coils and armature, and is then discharged from the commutator end of the generator.

Twelve GE model 752 series-wound, 4-pole, traction motors will power the four 3-axle fully equalized trucks under the two units. Each generator powers three motors on one track and has its own control and excitation equipment. This permits isolating any generator without affecting the remaining generators. Traction motors are permanently connected in parallel and have one-step field shunting. At the full-power rating of 8500 hp, the input to generator for each traction motor will be 708 hp. This compares with 562 hp on the present GTE locomotives. Axle gear ratio is 74/18, wheels 40 in. diam, maximum locomotive speed, 65 mph. Maximum continuous tractive effort is 146,000 lb at 18 mph. Three dynamic-braking grids give a peak braking effort of 147,000 lb at 22 mph.

Air from the traction-motor blower on each unit is carried to the motors by a main center-sill duct arrangement.

The 6-wheel trucks used on both units are new design with equal spacing between axles. Accessibility to traction motors is comparatively good for a three-motored truck. The bolster is

supported on four rubber sandwiches. Hydraulic shock absorbers dampen bolster side movements and vertical movements between equalizers and side frames.

Turbine load is applied by a 20-notch throttle which increases excitation of the traction generators. A Woodward governor supplies fuel in accordance with electrical load requirements unless overridden by the Hagan exhaust-temperature-control equipment. Exhaust-temperature control begins at 823 F. Over-temperature alarm is set for 860 F and over-temperature trip at 885 F. A notching guide in the cab indicates to the enginemen if surplus turbine power is available for increased electrical load.

Fuel will be carried in a 24,000-gal insulated fuel tender. These tenders, which will be furnished by the Union Pacific, are being converted from retired steam-locomotive tenders. Tenders will be provided with heater pipes for use of wayside steam at shops, and with 78-kw electrical heating capacity for emergency heating en route, using power from the diesel-engine generator.

The residual-fuel-supply pipe between the turbine and the fuel tender is tracer heated by waste air discharged from the atomizing air-dirt separator. This air has a temperature of approximately 500 F. When starting up cold, the turbine will operate on diesel start fuel until the supply pipe has been heated sufficiently by the waste atomizing air to permit the flow of heavy fuel.

Over-all length of the A and B units is 132 ft 6 in. Over-all length of locomotive and tender is 178 ft $11\frac{1}{4}$ in. with one type of tender, and 180 ft $7\frac{3}{8}$ in. with another type of tender. Fully loaded weight of the A and B units is 840,000 lb. The two units presented almost opposite structural-design problems. In the A unit it was necessary to overdesign the structural members in order to obtain the 80,000 lb of additional weight required for adhesion purposes. With the B unit it was necessary to use high-tensile-steel structural members in order to stay within the weight limitation of 70,000 lb per axle.

U. S. Army 30-Ton Gas-Turbine Locomotive

In 1952, the Transportation Corps of the U.S. Army established a project to determine how the unique characteristics of the gas-turbine engine might be applied advantageously to military railway motive power. In order to gain practical experience in the design and operation of this type of power, a small, lightweight, experimental locomotive was built, Fig. 7. This locomotive is a 30-ton, standard-gage unit with a 2-4-2 wheel arrangement.

The power plant of the 30-ton Gas-Turbine Locomotive consists of two Boeing Airplane Company model 502-2E gas-turbine engines which operate in parallel through a combining-reduction-gearbox. Power is transmitted through a three-speed planetary-gear transmission to the driving wheels, thereby providing the locomotive with three speed ranges in either direction of travel. Side-rod coupling is provided between the driving wheels. To facilitate maintenance, the complete power plant is mounted in the locomotive as a unit.

The Boeing Model 502-2E engine, as used in this locomotive, is of the free-power-turbine type and is rated at 150 hp at standard atmospheric conditions. The engine consists of a centrifugal compressor coupled to a single-stage axial-flow turbine, two combustion chambers, an accessory-drive section, axial-flow power turbine, and reduction gearing.

Control of either engine is accomplished by means of a governor which controls the speed of the gas-producer section. The governor, in turn, is controlled by the operator's lever. The engines as installed are designed to operate on diesel fuel.

Power from the two engines is combined to drive the three-speed transmission at reduced speed through a combining-reduction gear. The output of each engine is coupled to the combining gear through an overrunning clutch which allows single-engine operation of the power package. An air compressor for the brakes and control air is driven off the output of the combining gear. Mounted behind the engines and above the three-speed transmission is the lubricating-oil-cooling assembly which consists of four separate radiators for cooling the engine oil, combining-gear oil, and three-speed-transmission oil.

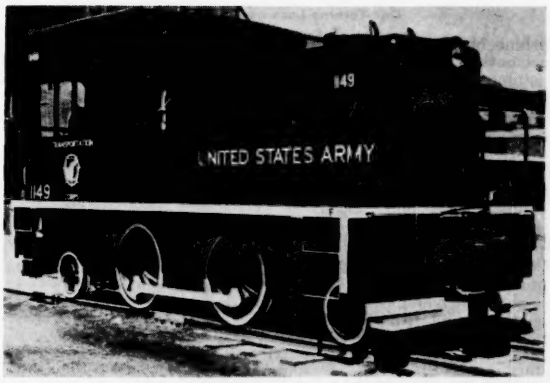


Fig. 7 U. S. Army 30-ton gas-turbine mechanical-drive locomotive

The chassis of the locomotive, which was built by the Davenport-Besler Corporation, is completely conventional and incorporates all standard equipment, such as air brakes, sanding equipment, and so forth.

The three-speed transmission consists primarily of a compound planetary-gear train, which is in constant mesh, and provides, through the action of hydraulically actuated clutches, gear ratios of 4.4:1, 2.33:1, and 1:1. Positioning of the ratio selector, which controls the hydraulic actuation, is done by a four-position pneumatic cylinder with manual selection by the operator.

The driving wheels are powered through an axle-hung transmission which serves as a reversing transmission, thus providing three speed ranges in either direction of travel. This transmission is shifted by a two-position pneumatic cylinder with manual selection.

This locomotive was accepted and delivered on September 23, 1954, and then operated in general switching service at Rock Island, Ill., until January, 1955. This first phase of the testing constituted a mechanical shakedown of the locomotive and provided preliminary test data of the over-all performance.

In February, 1955, the locomotive was transferred to Fort Eustis, Va., for extended engineering testing. In general the locomotive has performed well. The most serious trouble has been with the power-transmission components. Several failures were experienced because of the unanticipated high values of torque encountered when the three-speed transmission was shifted.

In spite of the transmission difficulties, there have been several significant results. Test operation showed that the twin-engine concept results in a lower over-all fuel consumption for a properly utilized locomotive of this type. For switchers, which have a relatively low load factor, the operation of only one engine when hauling light loads, and the elimination of idling by shutting down for even very short slack periods does save fuel. The ease with which these engines may be started (approximately 8-sec starting time) makes this operation feasible.

Another feature of the gas-turbine mechanical-drive combination is the lack of any continuous-rating restriction. This is a particular advantage for switching operations. The tractive effort of this locomotive was limited at low speeds, only by the wheel-to-rail adhesion. It could pull any load it could start, without time limitation.

At the time of this writing, test operation has been discontinued, and final disposition is undetermined.

Gas-Turbine-Locomotive Data

Turbine builder	Boeing Airplane Company
Locomotive builder	Davenport-Besler Corporation
Customer	Transportation Corps, U. S. Army
Order placed	March, 1953
Date first operated	September 23, 1954
Manufacturing rating—locomotive, hp	300
Number turbines per locomotive	2
Turbine-output shaft speed, maximum rpm	2750
Hp per ton locomotive weight	10
Wheel arrangement	2-4-2
Total weight, lb	60,000
Weight on drivers, lb	50,000
Length over-all, ft and in.	24-6
Wheel base, total, ft and in.	16-8
Wheel base, rigid, ft and in.	6-0
Width, ft and in.	9-8
Height, ft and in.	11-2
Service	Switching
Maximum speed, mph	35
Starting tractive effort, lb	14,000 (limited by adhesion)

Unit Data

Maximum rating, hp	150
Cycle	Simple open
Cycle arrangement	Split turbine
Top cycle temperature, deg F	1125
Pressure ratio	2.8
Weight, turbine unit, lb	240
Specific weight, lb per hp	1.6
Length, in.	39.89
Width, in.	23.20
Height, in.	24.43
Starting method	Electrical (batteries)
Air flow, lb per sec	3.5
Specific power, hp per lb per sec	43
SFC, lb per hp-hr (full load)	1.40
Combustors	
Type	Can
Number per engine	2
Fuel	Diesel (MIL-F-896 Class 1 or 2)
Compressor	
Type	Centrifugal
Number of stages	1
Speed, rpm (maximum)	35,000
Compressor turbine	
Type	Axial
Number of stages	1
Speed, rpm	35,000
Power turbine	
Type	Axial (free)
Number of stages	1
Speed, rpm (maximum)	23,700

Free-Piston Gasifier and Turbine-Electric Locomotive

Electro-Motive Division of General Motors advises under date of December 3, 1957, that the free-piston engine and turbine has been under development at their plant for the past $2^1/_2$ years. The engine used in the major portion of this development was a 1000-hp, single-cylinder, free-piston engine. EMD has designed a 2000-hp twin-cylinder free-piston engine which is now in operation. The manufacturer now feels that the state of development of the engine warrants a field application and intends to build an experimental locomotive using the 2000-hp twin-cylinder free-piston engine. The locomotive will have one GM-214 free-piston engine with a matching gas turbine, a gear box, a standard traction generator, traction motors, and control. It is intended that this unit will make possible a complete evaluation of this new prime mover in railroad motive-power service. The free-piston engine will burn the lowest cost fuels obtainable.

Pending construction and testing, no other details were released by Electro-Motive.

British Railways Gas-Turbine Locomotive

No. 18000. The British Railways gas-turbine locomotive No. 18000[8] which was built by Brown Boveri, continues to operate on the Western Region between London and Bristol, averaging about 2700 miles per week. The locomotive received a general overhaul in 1956, on which occasion the first row of the fixed blading of the turbine was renewed for the second time, and a new finned-type of liner was fitted to the combustion chamber and is giving good service.

The total mileage run by this locomotive as of the end of 1957 was 281,000, involving 7445 turbine hours.

No. 18100. Metropolitan Vickers 3000-hp locomotive, British Railways No. 18100, was the first British-built gas-turbine locomotive, and was delivered to the Western Region of British Railways in December, 1951. The locomotive was designed and built by Metropolitan Vickers as a joint project with the former Great Western Railway under an agreement which was continued by British Railways upon nationalization.

At the time the locomotive was delivered, it was the most powerful locomotive in use in Britain and had the lowest weight ratio in lb per hp of any self-propelled locomotive in main-line service. It was designed to handle 18 passenger cars with a total trailing load of 730 tons, and over a maximum grade of $^1/_{36}$ on the main line between London and Plymouth.

The locomotive completed 102,600 miles and 3370 turbine hr by September, 1953, at which time it was returned to the builders for some minor changes and conversion from distillate to residual-oil burning.

While the locomotive gave generally satisfactory performance in service, it showed little or no advantage in fuel costs compared with steam locomotives under the operating conditions involved. The fuel conversion was aimed to improve economy, but the modifications proved to be impracticable without substantial enlargement of the existing compact distillate-fuel combustion system. The consequent changes to the structure of the turbine and locomotive would have involved cost and delay, and these were deemed incompatible with the limited prospects for the early application of gas-turbine traction to the railways of Britain and of territories traditionally served by British manufacturers. The gas-turbine and generator unit has, therefore, been removed from the locomotive and it is being converted into an advance-training unit in connection with the new 50-cycle a-c electrification of sections of the London Midland and Eastern Regions of British Railways.

Coal-Burning Gas-Turbine Locomotive

In collaboration with the Ministry of Power and British Railways, C. A. Parsons & Company Ltd., and the North British Locomotive Company have in hand the development and construction of a coal-burning gas-turbine locomotive of 1750 hp.[9] This makes use of the exhaust heated cycle in which the compressed air is heated in a large heat exchanger and the pulverized-coal fuel is burned in the turbine exhaust. This system has the advantage of supplying clean air to the turbine. The maximum temperature at the inlet to the heat exchanger is 1560 F. The

[8] The table of particulars on this locomotive, as given in the previous ASME report, should be amended as follows: (*a*) Hp per 2000 lb weight locomotive, 19.34 (instead of 18.8); (*b*) total weight, 258,048 lb (instead of 266,890); (*c*) weight on drivers, 174,000 lb (instead of 177,920); (*d*) fuel, 950 Redwood sec (instead of 800).

[9] The revised particulars and estimated figures for this locomotive are set out on the attached table and replace those quoted in the previous report. Fig. 4 of the earlier report is now obsolete and should be omitted.

turbine is of the 2-shaft type, in which the high-pressure turbine drives the axial compressor, and the independent low-pressure turbine drives the locomotive through a gearbox and mechanical transmission.

At the present time the power unit is undergoing rig tests while mounted on the locomotive chassis, following the satisfactory outcome of which the North British Locomotive Company will carry out the final assembly of the locomotive.

While the original intention was to carry the locomotive on two 6-wheeled trucks, the increased weights of components have made necessary an additional pair of carrying wheels at the outer end of each truck, in order to meet the axle-load limitation required by British Railways.

C. A. Parsons-N.B.L. Coal-Burning Gas-Turbine Locomotive Revised Table of Particulars

Turbine builder	C. A. Parsons & Company
Locomotive builder	North British Locomotive Company
Electrical builder	None
Customer	Ministry of Power
Order placed	1951
Date first operated	...
Manufacturer's rating, locomotive, hp	1750
Number turbines per locomotive	1
Number generators per turbine	Mechanical drive
Generator rpm	2-speed gearbox
Hp per 2000 lb weight locomotive	1.045
Wheel arrangement	1A1A–A1A1
Total weight lb	335,000 lb (est)
Weight on drivers, lb	177,000 lb (est)
Length over-all ft and in	68-0
Wheel base, total ft and in	60-0
Wheel base, rigid ft and in	12-3
Width, ft and in	8-6¾
Height, ft and in	12-11
Service	Mixed
Maximum mph	50 (low) 72 (high)
Continuous tractive effort, lb	37,400 : 26,400
Mph at continuous tractive effort, lb, unit data	...
Rating to main generator for traction, hp	...
Maximum total rating, hp	1750
Cycle	Open[a]
Cycle arrangement	Split turbine
Top cycle temperature, deg F	1290 to turbine
Pressure ratio	...
Weight, turbine unit	...
Weight, lb per hp (number generated)	...
Length, ft and in	About 30
Width, ft and in	About 8
Height, ft and in	About 11
Starting method	Diesel engine
Air flow, lb per sec	...
Specific power hp per lb per sec	...
S.F.C. lb per hp per hr (full load) per cent (HHV) (full load)	...
HHV (of fuel used) Btu per lb	...
Combustors	
Type	Very large
Number	1
Fuel	Coal
Regenerator	
Type	Crossflow
Number	1, 1560 F maximum
Effectiveness, per cent	...
Compressor	
Type	Axial
Number stages	
Rpm	
Adiabatic efficiency, per cent	
Discharge, psia	
Main turbine	
Type	Axial
No stages	
Rpm	
Adiabatic efficiency, per cent	
Type cooling	
Power turbine (if separate)	
Type	Axial
No stages	
Rpm	8000
Adiabatic efficiency, per cent	
Type cooling	

[a] With exhaust heater.

Oil-Fired Gas-Turbine Locomotive. The English Electric Company have under development, as a private venture, an oil-fired, direct-drive, gas-turbine locomotive of 2700 hp which, through the co-operation of the British Transport Commission, has recently been successfully tested at the Locomotive Testing Station in Rugby, England.

Running time has been given to mechanical proving tests and to the establishment of performance and fuel consumption at various rates of working, in what are believed to be the first bench tests of a main-line locomotive prior to final completion.

As a result of the tests, the locomotive is to be completed for track trials.

Soviet Gas-Turbine Locomotive

Following the original development of gas-turbine locomotives in Switzerland and their subsequent introduction in Great Britain and in the U. S., and being impressed by the success of the locomotives used for heavy long-distance work on the Union Pacific Railroad, the Soviet industry has recently embarked on a program of gas-turbine locomotive design with a view to possible production. While at present experimental units incorporating widely divergent design features are being considered, it is becoming clearer that the emphasis may ultimately be directed to the use of solid fuel, particularly in view of a widespread program of diesel working and electrification. Also, a recent Party resolution was to end steam-locomotive building at an early date. By 1960 the number of electric locomotives planned calls for an increase of 2000, and some 2250 two-unit diesel-electric locomotives are also planned. The first gas-turbine locomotives are scheduled to be in service by that time also.

Kharkov 6500-hp Type. The first designs, by the Kharkov locomotive works, envisage straightforward open-cycle locomotives consisting of two Co-Co units developing a total of 5280-hp at the drawbar, with a layout as shown in Fig. 8. The general design of each unit is somewhat reminiscent of the Brown Boveri and Metropolitan-Vickers locomotives developed for British Railways. Each unit is powered by a single gas turbine of 3220 hp, developed at 7000 rpm, and a gas inlet temperature of 700 C. The number of stages is 12 for the compressor and five for the turbine, pressure ratio being 5:1. The two generators are gear driven and their maximum speed is limited to 1600 rpm. All axles are driven by nose-suspended motors, but the rims of the axle-mounted gear wheels are connected to the disks via springs dealing with purely tangential load, this being the standard practice with most Soviet electric and diesel-electric locomotives. The locomotive design incorporates a heat exchanger having an efficiency of 50 per cent, and over-all locomotive thermal efficiency is estimated at 21.3 per cent. Each unit is estimated to weigh 137 tons, which for the 41½-in-diam wheels results in an axle-load/wheel-diameter ratio of 6.62 a high value compared with West European practice.

Kolomna 7500-hp Design. Another design along somewhat similar lines was evolved by the Kolomna works. Here again a two-unit locomotive is envisaged, this time powered by two 3740-hp turbines and developing 5400 hp at the drawbar. The speed of the gas turbine has been increased to 8500 rpm and the gas temperature has been raised to 727 C. The compression ratio has been raised to 6:1, and though the number of turbine stages has

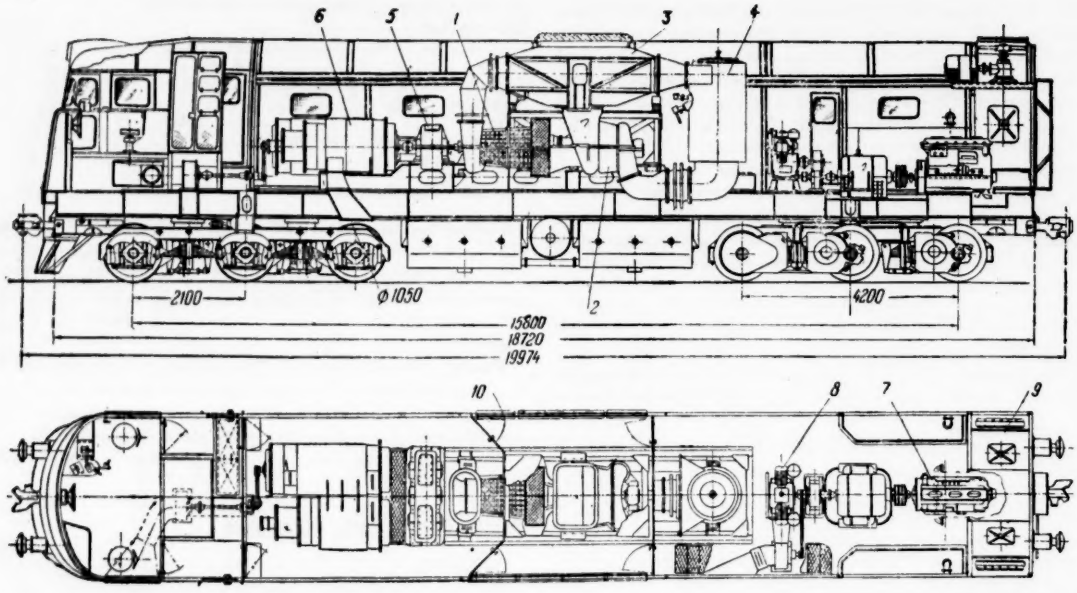


Fig. 8 Gas-turbine-locomotive design by the Kharkov Locomotive Works: 1 Compressor, 2 turbine, 3 heat exchanger, 4 combustion chamber, 5 reduction gear, 6 generator, 7 auxiliary diesel engine, 8 brake compressor, 9 oil cooler, 10 air filter

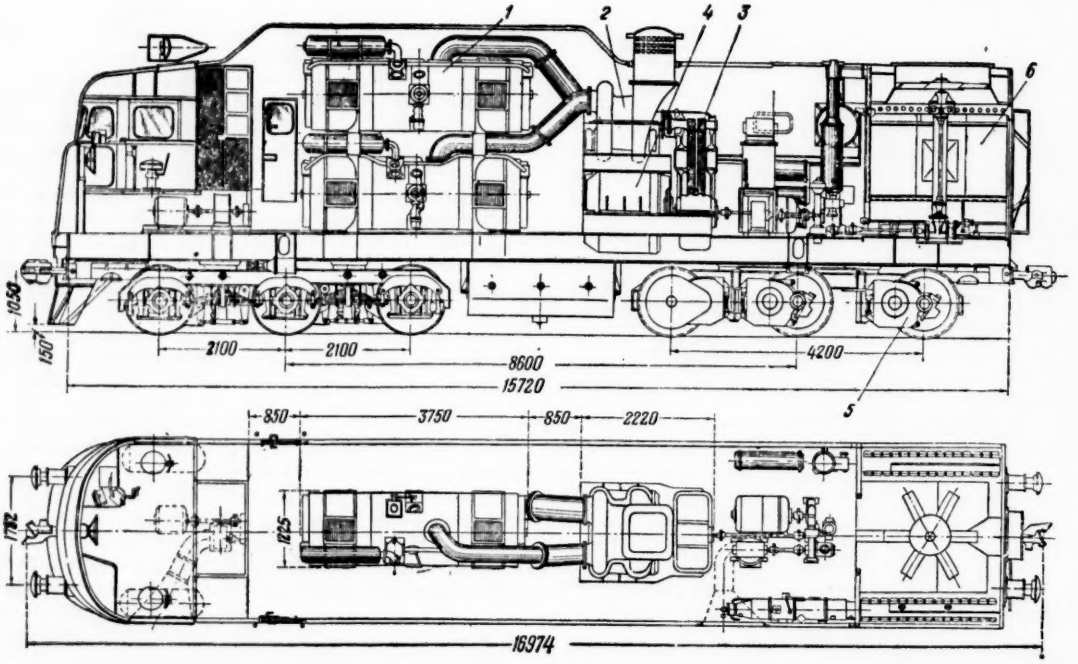


Fig. 9 Free-piston gas-turbine-locomotive design by Kharkov Locomotive Works: 1 Free-piston gas generator, 2 gas turbine, 3 reduction gear, 4 generator, 5 traction motors

been reduced to four that of the compressor remains unaltered at 12. No heat exchanger is to be used, but because of the improved thermal characteristics the envisaged over-all efficiency of the power plant is estimated at 20.8 per cent. Both locomotives are designed for heavy-goods traffic, and because of this the maximum speed is limited to approximately 100 km per hr, or 62 mph.

Gas-Generator Units. The development of free-piston gas generators and the possibility of their use with locomotives has also received considerable attention, particularly since a number of free-piston generators are now running in the USSR. Two locomotive designs not dissimilar to one another and incorporating free-piston generators have been evolved by the Voroshilovgrad and Kharkov locomotive works, and a third has been built in that country. Voroshilovgrad has also built an experimental gas generator for the application concerned. Both designs envisage locomotives consisting of two Co-Co units developing a total of 6000 hp, and one unit of the Kharkov design is shown in Fig. 9. Here two gas generators are installed, one above the other, feeding a common turbine running at a maximum speed of 4800 rpm with a gas temperature of 444 C. The weight of one locomotive unit is 125 tons. In the case of the Voroshilovgrad design each unit is powered by four gas generators which feed gases at a maximum temperature of 462 C to a common turbine running at 7000 rpm max, with estimated over-all efficiency of 29 per cent. The estimated weight of one unit is 142.5 tons, which results in an axle-weight/wheel-diameter ratio of 6.9. Here again maximum speed is limited to about 60 mph.

Coal-Burning Considerations. In view of the difficulties associated with the operation of direct coal-burning gas turbines it was only natural that the designs aiming at the use of coal were in the first instance directed to the development of locomotives incorporating indirect combustion, that is, the burning of coal and transmission of heat to the turbine via a primary heat exchanger or air heater. However, present materials limit the air-heater temperature to about 700 C. The consequent temperature of the air before the turbine would scarcely exceed 600 C limiting the over-all power-plant efficiency to about 10 or 11 per cent, which is not sufficiently high to justify the development of this type of locomotive. Development of automatic part-load control also presents considerable difficulty. To overcome these shortcomings N. I. Belokon suggested the use of afterburning. In this idea a combustion chamber utilizing liquid fuel is arranged downstream of the air boiler and the temperature of the air preheated in the boiler to some 575 or 600 C is increased to 720 or 750 C on its passage through the secondary combustion chamber. Since the amount of liquid fuel used comprises about 20 to 30 per cent of the total, the gases upstream of the turbine contain only about 1 per cent of the products of combustion, and so can be considered as being practically equivalent to pure air. Because of this they can be utilized for combustion in the primary chamber. Here part of the turbine exhaust is led to the combustion chamber, while the large volume is fed into the boiler downstream of the primary combustion chamber, where it is utilized to cool the primary gases before entering into the heat-exchanger part of the boiler, which is approached at a temperature of about 1000 C.

A Bo-Bo-Bo-Bo single-underframe locomotive based on the utilization of fuel along these lines has been designed by the Railway Research Institute. It is proposed for 3000 hp and the distribution of the equipment is as follows. The auxiliary diesel engine driving the generator and brake compressor is mounted under the end driving cab, to the rear of which is a coal bunker holding 8 to 10 tons of fuel, which should be sufficient for 5 to 6 hr of running. The rear end of the locomotive contains the power plant, secondary combustion chamber, secondary fuel supply, oil cooler, and batteries. The locomotive is mounted on four bogies, but only three of these are powered, no motors being provided in the leading bogie. The total weight, including two-thirds fuel supply, is estimated at 173 tons.

Air Heater. The cornerstone of the whole design is the air heater. This consists of three parts: (*a*) The combustion chamber; (*b*) the heat exchanger proper, made of high-temperature alloys; and (*c*) the outlet section, made from ordinary steel sheets. The combustion chamber is cylindrical in shape and is suitably insulated on the inside. To cool the combustion-chamber walls the air leaving the compressor is forced through the tubes and then enters the tail end of the tubular heat-exchanger. This division of the boiler into two tube sections reduces the amount of alloy steel required for its construction. In accordance with the design data the locomotive should utilize some 70 to 80 per cent of solid fuel and only 20 to 30 per cent of liquid fuel, the over-all power-plant efficiency being 14.5 per cent at maximum output. The advantages expected from this design are good part-load efficiencies and a greater elasticity in traffic ensured by the use of the secondary combustion chamber.

However, a considerable amount of research work is being carried out on direct coal-burning gas turbines by a number of scientific research establishments, which, in collaboration with the Kharkov locomotive works, have prepared a design of a Bo-Bo-Bo-Bo locomotive, the power plant of which is similar to that of the gas-turbine locomotive shown in Fig. 8. The additional space required by the coal storage and the processing and gas-cleaning equipment have, however, increased the length of the locomotive by about 20 ft.

Swiss Gas-Turbine-Electric Locomotive

As stated in the 1952 report, the 2200-hp gas-turbine-electric pioneer locomotive No. 1101, built by Brown Boveri & Company, Ltd., Baden, Switzerland, and Swiss Locomotive and Machine Works, Winterthur, Switzerland, operated a total of 237,000 miles from 1941 until 1954 over French, German, and Swiss railways. It was then taken out of service since its prototype purpose had been successfully fulfilled. Furthermore, practically the whole network of Swiss Federal Railways is electrified, and a locomotive requiring imported fuel oil is not desirable. It was planned to convert this locomotive to a straight electric locomotive with rectifier conversion equipment for experimental use on a 50-cycle commercial-frequency railway electrification. Brown Boveri advised under date of December 13, 1957, that the rebuilding of the locomotive will be delayed for several unspecified reasons.

Czechoslovak State Railway Locomotive

In certain parts of Czechoslovakia, railway lines exist where the problem of electrification is considered too difficult because of landscape configuration, extensive grades, and large altitude variations. The Czechoslovak State Railway Authorities, in realizing that the existing steam locomotives which serve these lines would not be able to deal with the growing flow of traffic, expressed the need for another type of locomotive which would suitably supplement the present rolling stock.

As a result of this demand, and following a program of systematic research and development at both the manufacturing works and the state research institutes, a locomotive powered by a gas-turbine engine was constructed by the V. I. Lenin Works (formerly the Skoda Works) at Plzeň.

The oil-burning single-unit mechanical-drive locomotive, which develops a maximum output of 3200 hp, is of the Co-Co type and weighs 123 tons when fully equipped. It can be used to haul either freight trains at a maximum speed of 50 mph or express passenger trains at a maximum speed of 77.6 mph. The prototype of the locomotive was tested on the State Railways by the

Fig. 10 3000-bhp gas-generator locomotive, USSR

end of January, 1958, and the first trials hauling a freight train of 960 tons have proved its economy in practice.

The gas turbine, a two-shaft, open-cycle arrangement, comprises an axial-flow compressor driven by a turbine, a combustion chamber, a free, single-stage turbine (providing the output power), a heat exchanger, and an auxiliary starting engine.

The starting engine, an air-cooled Tatra T IIIA diesel unit, is connected to the compressor through a gearbox, and a sliding coupling, which automatically disengages the engine when the compressor reaches self-sustaining speed.

Power is transmitted to all six axles, that is, two bogies each incorporating three axles, via a mechanical drive system; this is arranged above the main frame, passing down through the central pivot points of the bogies. The fuel tanks and the lubrication and regulation oil tanks are located beneath the main frame of the locomotive. The fuel burned in the combustion chamber comprises the wastes from the refining of crude petroleum and the production of artificial kerosene.

For movement of the solo locomotive in both directions, in and around the railway station at 7.5 mph, sufficient power is provided by the Diesel engine. The locomotive can be operated from the front or rear cab, one driver having complete control.

Advantages claimed for the turbine-driven locomotive are improved thermal efficiency, that is, at least twice that of most modern steam counterparts, increased output, and economy resulting from decreased ambient air temperature, and ability to operate without water consumption.

Thus the turbine-driven locomotive represents a significant advance in the development of locomotives in Czechoslovakia, and further tests are now in progress on the lines of the Czechoslovak State Railways in an attempt to speed up the production of the first prototype series.

Swedish State Railways Locomotive

The 1000-hp mechanical-drive diesel-turbine locomotive built by Aktiebolaget Gotaverken of Goteborg, Sweden, has been in service since February, 1955, and operates daily on a 244-mile-round-trip local passenger run. The power plant consists of a 5-cyl opposed-piston two-cycle supercharged diesel engine which supplies gas at a temperature of 935 F to a 7-stage axial-flow gas turbine. By means of a change-over valve, gas does not pass through the turbine when the diesel engine is idling. The mechanical transmission consists of clutches, double-reduction gearing, and side rods. An over-all efficiency at rail of 31.5 per cent with fuel consumption of 0.44 lb per rail hp-hr is claimed. Idling fuel consumption is 8 per cent of the full-load fuel consumption.

Under date of December 10, 1957, Aktiebolaget Gotaverken, designers of the diesel turbine machinery, advise that: "The locomotive has continued its operation in the same type of passenger service as the previous year. Certain difficulties have been experienced with the cast center sections of the cylinders in the opposed-piston engine, but these are expected to disappear when new combustion chambers of wrought steel are incorporated. Otherwise the machinery has performed up to expectations. The turbine blading keeps clean, and no attacks of any kind have been observed on the blading. The reverse-reduction gear with the hydraulically operated friction clutches for forward and reverse has not required any opening up since the delivery. The bearings of the high-speed pinion have been inspected and found in excellent condition.

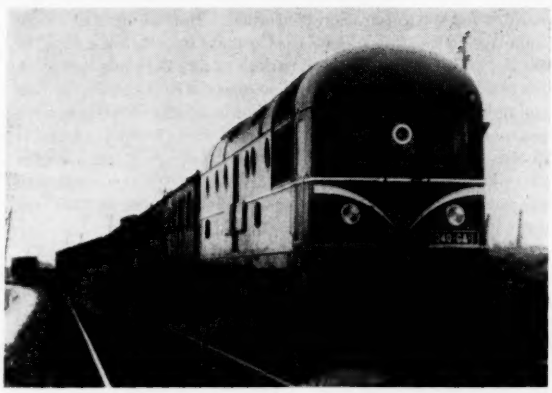


Fig. 11 1100-bhp Renault-Sigma gas-generator locomotive, France

"The maneuverability has proven excellent and the simplicity of the control system has facilitated the training of personnel. The exhaust is invisible over the entire range of operation from idling to full load. Owing to the fact that the driving wheels are coupled by side rods, spinning is seldom experienced even during bad-weather conditions."

Gas-Generator Locomotive

The gas-generator locomotive is not a new invention. The first one ran in 1932-1933, and though progress in the ensuing period has been over-shadowed by diesel activities, nevertheless the gas-generator locomotive has given satisfactory results and has become practical on a small scale. The theoretical conception of the gas generator is older still. Gotaverken and Alfred J. Buchi both had ideas of this kind by the early 1920s. The power portion of a gas-generator locomotive comprises a diesel engine, an exhaust-gas turbine, and a transmission system. Basic details are given in the paper on compound piston and turbine engines.

With only half-a-dozen or so gas-generator locomotives built, one can be impressed by the regular and satisfactory performance in France, on normal revenue trains, of the Renault-Sigma loco-

Table 1 Gas-generator locomotives

Item No.	Builder	Country	Year	B.h.p. (1)	Wheel arrgmt.	Type of trans-mission	Weight in working order, tons	Weight in lb. per b.h.p.	Max. axle load, tons	Top speed, m.p.h.	Rail tractive effort at top speed, lb.	Starting tractive effort, lb.
1	Gotaverken-Nohab	Sweden	1933	650	1-B-1	Mech.	53	183	14	45	4,300	13,500
2	Renault-Sigma	France	1952	1,100	B-B	Mech.	53	108	13¼	77	3,300	25,000
3	Soviet works	U.S.S.R.	1954	3,000	3(Bo-Bo)	Elect.	200 (2)	150	18½	62	?	75,000
4	Gotaverken-Motala	Sweden	1955	1,300 (4)	1-C-1	Mech.	63(5)	108	13½	56	6 600	30,000
5	Renault-AFL-Sigma	France	U.C. in 1956	1,100	B-B	Mech.	51 (3)	103	13	78	3,500	24,000
6	Renault-AFL-Sigma	France	U.C. in 1956	2,200	C-C	Mech.	106 (3)	108	18	78	7,750	64,000
7	Lima Hamilton-Westinghouse	U.S.A.	U.C. 1951-52	3,000	?	Elect.	Construction stopped on fusion of Lima Hamilton with the Baldwin Locomotive Works					

(1) Figures in this column are the gas-generator h.p., plus the auxiliaries where those are driven separately, so that the figure is equivalent to that of the b.h.p. of a diesel locomotive, and is given so that comparisons can be made with diesel locomotives; in general descriptions it is often the turbine output of gas-generator locomotives which is given, e.g. the Renault-Sigma locomotive (item 2) is often spoken of as 1,000 h.p.
(2) Approximate. (3) Estimated. (4) In winter a 170-h.p. alternator for train heating is driven additionally, making the equivalent of 1,470 b.h.p. (5) Including electric train-heating equipment

Table 2 Comparison on basis of approximately equal weights of gas-generator and diesel locomotives

Item No.	Type of locomotive	Locomotive	Wheel arrgmt.	Trans-mission	Working order weight, tons	Max. axle load, tons	B.h.p.	Weight per b.h.p., lb.	Top speed, m.p.h.	Rail tractive effort at top speed, lb.	Continuous rated tractive effort (lb.) and corresponding speed (m.p.h.)	Starting tractive effort, lb.	Gauge
1	Diesel	Tunisian Rlys (Alsthom)	Bo-Bo	Elect.	52	13·0	610	190	45	3,250	16,500 at 8¼	30,000	Metre
2	Diesel	Gold Coast Rlys. (English Electric)	Bo-Bo	Elect.	53	13·5	750	158	55	2,600	19,900 at 9·6 one hour	30,000	3ft. 6in.
3	Diesel	New Zealand Govt. Rlys. (English Electric)	Bo-Bo	Elect.	52	13·0	660	176	55	1,900	12,700 at 15½	29,000	3ft. 6in.
4	Diesel	Great Northern Rly. of Ireland (M.A.K.)	D	Hydr.	55	14·0	800	154	55	3,550 (1)	20,000 at 9 (1)	23,500(1)	5ft. 3in.
5	Diesel	Dakar-Niger Rly. (A.F.L.)	B-B	Hydr.	50	12·75	850	132	44	?	?	30,000	Metre
6	Diesel	S.N.C.F. (Renault)	B-B	Mech.	52	13·0	840	137	62	?	?	30,000	4ft. 8½in.
7	Diesel	Netherlands Rlys. (Alsthom)	Bo-Bo	Elect.	59	15·0	925	143	50	5,500	21 800 at 14½	35,000	4ft. 8½in.
8	Diesel	Argentine National (Werkspoor)	Bo-Bo	Elect.	59	15·0	600	220	47	2,250	12,000 at 13½	23,600	5ft. 6in.
9	Diesel	Westphalian Rly. (Henschel)	B-B	Hydr.	59	15·0	875	150	56	2,200 (1)	21,000 at 8½ (1)	28,000(1)	4ft. 8½in.
10	Gas-generator	Swedish State (Gotaverken)	1-C-1	Mech.	63	13·5	1,300	108	56	6,600	—	30,000	4ft. 8½in.
11	Gas-generator	S.N.C.F. (Renault-Sigma)	B-B	Mech.	53	13·25	1,100	108	77	3,300	—	25,000	4ft. 8½in.

(1) With top alternate gear engaged

Table 3 Comparison on basis of approximately equal bhp of gas-generator and diesel locomotives

Item No.	Type of locomotive	Railway (Builder)	Wheel arrgmt.	Trans-mission	B.h.p.	Working order weight, tons	Max. axle load, tons	Weight per b.h.p., lb.	Top speed, m.p.h.	Rail tractive effort at top speed, lb.	Continuous rated tractive effort (lb.) and corresponding speed (m.p.h.)	Starting tractive effort, lb.	Gauge
1	Diesel	C.I.E. (B.R.C.W.)	A1A-A1A	Elect.	1,000	75	14·0	168	75	3,000	16,900 at 16	41,000	5ft. 3in.
2	Diesel	Sierra Leone Dev. Corp. (B.R.C.W.)	A1A-A1A	Elect.	1,050	72	13·0	153	50	4,400	15,500 at 15	30,000	3ft. 6in.
3	Diesel	Algerian Rlys. (De Dietrich)	Co-Co	Elect.	1,070	71	12·0	148	53	4,400	22,000 at 9½	38,000	3ft. 6in.
4	Diesel	Japanese National (Mitsubishi)	Bo-Bo	Elect.	1,000	62	15·75	138	56	4,000	14,300 at 18	40,000	3ft. 6in.
5	Diesel	German Federal (Krauss Maffei)	B-B	Hydr.	1,000	59	15·0	132	62	4,400	27 000 at 10	39,000	4ft. 8½in.
6	Gas-generator	S.N.C.F. (Renault-Sigma)	B-B	Mech.	1,100	53	13·25	108	77	3,300	—	25,000	4ft. 8½in.
7	Diesel	C.I.E. (Metro-Vick)	Co-Co	Elect.	1,200	85	14·25	158	65	3,300	17,400 at 20½	50,000	5ft. 3in.
8	Diesel	Ceylon Govt. Rly. (Brush)	A1A-A1A	Elect.	1,250	87	15·0	155	56	3,750	24,600 at 11½	40,000	5ft. 6in.
9	Diesel	General Motors Standard G-12	Bo-Bo	Elect.	1,425	72	18·0	114	65	6,000	25,800 at 15½	47,000	4ft. 8½in.
10	Gas-generator	Swedish State (Gotaverken)	1-C-1	Mech.	1,300	63	13·5	108	56	6,600	—	30,000	4ft. 8½in.

motive for over four years, and also by the 2-yr service of the second Swedish locomotive. These are beyond the standards normally expected from trial or experimental units.

The Renault-Sigma locomotive delivers up to 910 hp at the rail. Allowing for transmission efficiency and turbine mechanical efficiency, its gas hp plus the power absorbed by the locomotive auxiliaries is about 1100. This is the hp which corresponds to the bhp usually given for diesel locomotives. The weight of this locomotive with all supplies on board is 53 tons, or 108 lb per bhp; but the unit gives a starting tractive effort in excess of 25,000 lb, 20,000-lb effort at 13 mph, and has a top speed of 77 mph, at which point 3300 lb of tractive effort are still available.

Tables 1, 2, and 3 show the characteristics of gas-generator locomotives built or under construction, and those of modern diesel locomotives of approximately equal power and equal weight. The data given are essentially comparable and are for line-service locomotives. The advantages of gas-generator locomotives in power-to-weight ratio, in maximum use of the installed hp, and in flexibility in operation are striking. In particular, the data relating to items 6 and 11 in Table 2 are interesting, in that both locomotives have the same length, the same trucks, the same pivot pitch, and the same superstructure frame, but the gas-generator locomotive gives 30 per cent more output than the diesel for 2 per cent more weight, though this diesel locomotive itself has quite a favorable power-to-weight ratio.

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MARINE

This is a progress report on the marine gas turbine and free-piston gas turbine during the 5-yr period, 1953–1957. Since the merchant and naval services encompass many similar operating conditions—in spite of many differing requirements—the entire marine field will be reviewed in a single paper. In the first report [18],[1] separate papers appeared on marine, merchant, and naval applications.

Open and closed-cycle gas-turbine, nuclear gas-turbine, and free-piston gas-turbine prime movers will be discussed along the following lines: Applications, operating experience, development, advantages, disadvantages, economics, and the future potential in the marine field.

THE increase since 1952 in number of marine gas-turbine applications has been encouraging. At that time only a small number of ship and boat installations had been completed. Many of these were experimental. The developmental types were essential in order to obtain installation and sea experience. Early marine plants include such well-known ships and boats as: (*a*) *MGB 2009* (British Navy); (*b*) *Torquil* Launch (British); (*c*) Plane Personnel Boat, 24-ft (U. S. Navy); (*d*) *Bold Pathfinder* MTB 5720 (British Navy); (*e*) *Bold Pioneer* MTB 5701 (British Navy); (*f*) *USS Timmerman* (DD828) (U. S. Navy); (*g*) *Auris* (Shell Petroleum Company Tanker), Fig. 1; (*h*) MSB-5 Class Minesweeper (U. S. Navy).

Published material relating to marine gas turbines and free-piston gas turbines has been reviewed in recent papers [1, 2]. By the end of 1957, the picture of total engines, total horsepower, and number of ships and boats involved had changed rapidly. A summary of the installations is presented in Table 1. The combined horsepower installed, or building for installation, is about 313,000—58,000 auxiliary and 255,000 propulsion. Free-piston gas turbines and gas turbines have 107,000 and 206,000 hp, respectively. In naval service the gas turbine has 175,000, the free piston 37,000 hp. In merchant use the gas turbine has 31,000, the free piston 70,000 hp. The number of naval and merchant vessels involved in these installations is 288 and 39, respectively. The total of gas-turbine engines is 460 and free-piston gas turbine 61.

Factors contributing to the rapid rise in the marine field include: (*a*) Improvements in engines; (*b*) actual engine operation at sea; (*c*) new application.

Improvements in engines have been made through use of new materials, higher component efficiencies, and better mechanical design. The development trend in engines is illustrated by the drop in specific fuel consumption (lb of fuel per hp-hr) of the Boeing engine, Fig. 2. In 1947 the engine specific was 1.8 lb per hp-hr. This was down to 1.0 in 1953. An extrapolation of the trend indicates that 0.7 specific may be reached in 1957–1958.

Accessory power and mechanical losses have a bearing on turbine performance and must be controlled. Compressor efficiency was improved. In 1950 a pressure ratio of 3:1 and 73 per cent efficiency were obtained. With additional development a 4.25:1 pressure ratio and a 77 per cent total-to-static efficiency resulted. Research toward combustion and burner improvement have resulted in longer liner life, elimination of carbon deposits, and reduction in burner noise. Metallurgy and design improvements contributed to solution of the serious problems encountered with the diaphragms, blades, warping, and thermal shock. Finally,

[1] Numbers in brackets designate References at end of paper.

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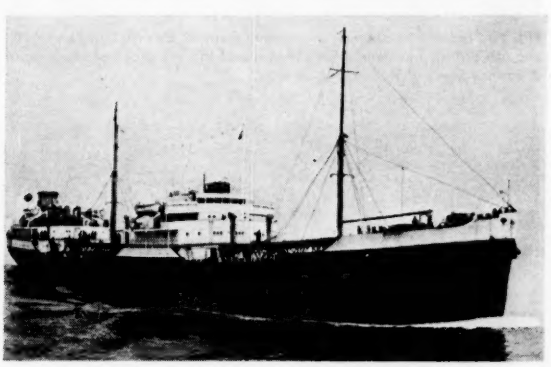


Fig. 1 ***Auris*** **shows off her gas turbine during demonstration run down the Solent, January, 1952 (Courtesy Shell Petroleum Company, Ltd.)**

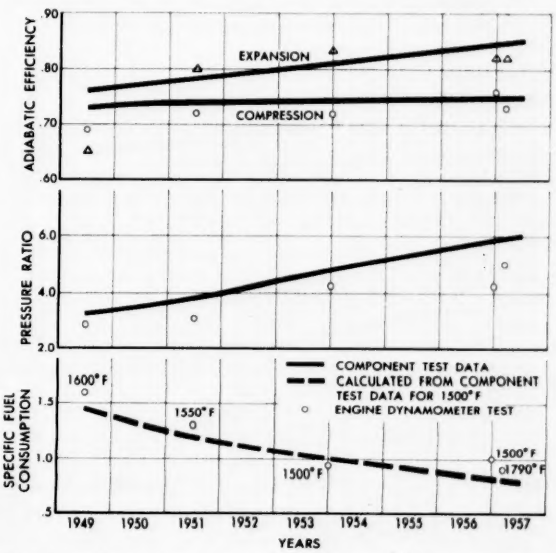


Fig. 2 Development trends in small gas turbines [3] (Courtesy Boeing Airplane Company)

Table 1 Summary of gas turbines and free-piston gas turbines in marine use[a]

Engine	Merchant			Naval			Total		
	Number ships	Number engines	Total hp	Number ships	Number engines	Total hp	Ship	Engines	Horse-power
Gas-turbine									
Propulsion	9	13	25000	61	80	124000	70	93	149000
Auxiliary	13	13	6000	206	354	51000	219	367	57000
Total	22	26	31000	267	434	175000	289	460	206000
Free-piston gas turbine									
Propulsion	15	17	69000	21	42	37000	36	59	106000
Auxiliary	2	2	1000	..	..	...	2	2	1000
Total	17	19	70000	21	42	37000	38	61	107000
Total									
Propulsion	24	30	94000	82	122	161000	106	152	255000
Auxiliary	15	15	7000	206	354	51000	221	369	58000
Total	39	45	101000	288	476	212000	327	521	313000

[a] Based on reviews of published material.

an engine of some 220 hp with an overhaul interval of 900 hr is available. This compares with the early production engine with a 250-hr overhaul period. This new engine is about the same package size as the early 100-hp unit [3].

The tendency to be overcautious about accepting new ideas is not unique in any field. The gas turbine was installed in both naval and merchant ships to get "experience" without, perhaps, consideration for the over-all economics of the particular installation.

Gas-turbine-powered aircraft are a *must* in military service. Commercial aviation is making great strides in replacing reciprocating engines with turboprops and turbojets. Certain locomotive service has been well suited for gas turbines. At this time the Union Pacific has 25 4500-hp gas-turbine-driven locomotives in use [4]. In addition, 45 8500-hp gas-turbine-driven locomotives are on order [5].

The requirements of marine propulsion have been mentioned by numerous writers. One grouping [6] shows, in order of importance, the following requirements for merchant ships:

1 Maximum reliability.
2 Minimum operating fuel cost.
3 Minimum first cost.
4 Minimum maintenance and operating-personnel cost.
5 Maximum maneuverability.
6 Minimum weight.
7 Minimum volume.

Sea experience with the gas turbine in the past several years has proved a number of the earlier claims. Those which are considered of greatest value and interest to the marine field are:

Reliability. This has been demonstrated in both merchant and naval service as the following examples illustrate:

1 The *Auris*, Shell Petroleum tanker, has had over 19,800-hr operating time on her 1200-hp B.T.H. gas turbine driving an alternator. She made an Atlantic crossing on gas-turbine power alone in March, 1952 [7].

2 The *John Sergeant*, Fig. 3, powered with a General Electric 6600-hp gas turbine has completed seven trouble-free round-trip Atlantic crossings up to April, 1958.

3 Emergency power is now being supplied on the *USS Gyatt* by a 300-kw generator set, Fig. 4, driven by a Solar Jupiter T-520 gas turbine, with a maximum rating of 500 hp which is brought up to speed with a combustion-air starter. Starting can be either manual or automatic [8]. The set has had 136 starts within a 13-month period, March, 1957–April, 1958.

Fewer Operating Personnel Required. The simplicity and automatic controls incorporated in the gas turbine have appreciably lowered operating costs. In four Liberty ships, converted from their original propulsion plants, engine-room crews are now:

Ship	Ship propulsion	Number in engine-room crew
Benjamin Chew	Steam turbine	16
Thomas Nelson	Diesel	14
John Sergeant	Gas turbine	12
William Patterson	Free piston	12

The experience of the Union Pacific Railroad shows that fewer enginemen and trainmen are required to operate gas-turbine

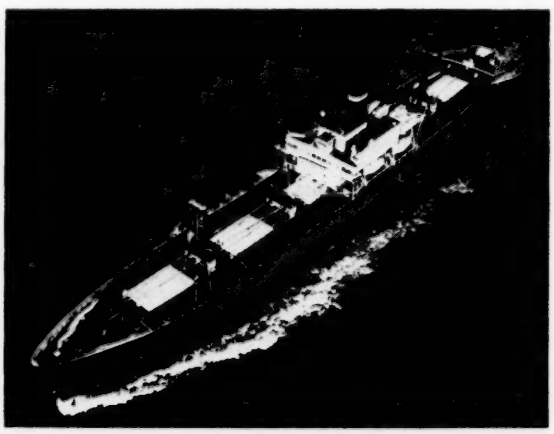


Fig. 3 The *John Sergeant* powered with General Electric Company 6600-shp gas turbine has completed seven round-trip crossings of the Atlantic (Courtesy General Electric Company)

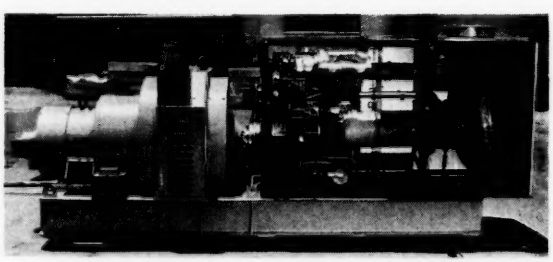


Fig. 4 The 300-kw gas-turbine-driven generator set on the *USS Gyatt* (Courtesy Solar Aircraft Company)

locomotives [4]. This indicates that the operating crew for the gas-turbine engine is about 25 per cent smaller than the diesel. While locomotive operating conditions differ from those of a ship, it is obvious that the size of an operating crew will be considerably less for a properly designed gas turbine than for either diesel or steam-propelled ship. "The reduction of one man per watch can result in a saving in operating cost comparable to an improvement in plant thermal efficiency of about 10 per cent on a cargo vessel with a propulsion plant rated at 20,000 hp (based on 5000 operating hr per yr under way and fuel costing $2.70 per bbl)" [6].

Rapid Starting. The gas turbine can be started from a cold condition and deliver full power within a relatively short period of time, varying with engine type and size. Several illustrations show the rapid starting ability of this prime mover: (a) Time for the Proteus marine engine, Fig. 5, Bristol Aero Engines, Ltd., rated at 3500 shp, is 30 sec [9]; (*b*) for the 300-kw emergency-generator set on the *USS Gyatt*, 10 sec [8]; (*c*) for the Metropolitan Vickers G-2, 4500-hp gas turbines on *HMS Bold Pioneer* idling speed is reached in about 30 sec and full power in another 12 sec [10].

Cold-Weather Reliability. The quick and reliable cold-weather starting abilities of the gas turbine are extremely important for auxiliaries, such as pumps and deicers, and small-boat propulsion and generator drives. A Boeing 160-hp engine, left on deck overnight in a temperature that ranged from 9 to 15 F started easily in tests [11].

Low Maintenance. Early experience with the *Auris* gas turbine showed a maintenance cost of 0.06 cent per kwhr as compared to diesel (British Diesel Engine Users Association) of 0.30 cent per kwhr and steam turbines, without boilers (British Electrical Authority) of 0.049 cent per kwhr [12]. A diesel engine in a gas-turbine plant showed a maintenance cost of 0.19 cent per kwhr, and a 5400-kw Brown-Boveri gas turbine using blast-furnace gas had a maintenance cost of 0.0069 cent [13] per kwhr. Experience on the Solar 500-hp Jupiter after a 1500-hr ship test on two engines indicates an overhaul cost of 5 to 10 per cent of the initial engine cost as compared to major overhaul of 30 to 40 per cent of new-engine cost for reciprocating engines [14].

Low Installation Cost. The compactness of even large gas turbines, such as the 6600-hp unit for the *John Sergeant*, permits mounting on an integral base which can contain all accessories and installation at lower cost than if the various parts were assembled separately [6]. The 300-kw emergency set now on the *USS Gyatt* was installed as a complete package.

Low or No Cooling Water Requirement. Heat rejection to lubricating oil is extremely low in all gas turbines. Large industrial-type turbines as well as marine plants require small quantities of cooling water as compared to other prime movers. With proper design, air can replace water for cooling. The Lycoming T-53 turbine has recently been built into a marine-propulsion package, Fig. 6. The turbine and gear-lubricating oil are cooled by a hydraulically driven fan which forces air through the oil-air heat exchanger [15].

Small Weight and Space. The simple-cycle gas turbine may occupy about 1/5 the volume and have 1/10 the weight of the diesel engine [18]. The following comparison indicates relative weights and volume of various type prime movers of equal power rating:

Type	Weight, lb	Volume, cu ft
Gas turbine	230	10.8
Aircraft, gasoline, reciprocating	356	17.6
Automotive, gasoline, reciprocating	1500	34.5
Diesel	2650	53.3

Specific weight and volume vary with the type plant and the service. Gas-turbine propulsion plants for high-speed naval craft may have specific weights of 2.3 lb per shp and a full power rating for 300 hr. The gas-turbine plant, turbine, gear, and regenerator in the *John Sergeant* have a specific weight of about 45 lb per shp [16].

Ease of Overhaul. The time required to disassemble, inspect, and overhaul gas turbines will vary with the ship installation as well as type and size of engine. In spite of crowded engine space which tends to increase removal and installation time, the Boeing 160-hp engine in the U. S. Navy MSB-5 Class Minesweep, can be removed and disassembled for inspection by operating personnel in 6 hr; and a crew of four can remove the four engines in 8 hr and install and have replacement engines operating in an additional 20 hr [17]. The in-port maintenance time for the 1200-hp gas turbine in the *Auris* is only a fraction of that necessary on a steam plant. When ". . . compared to the Diesel installation,

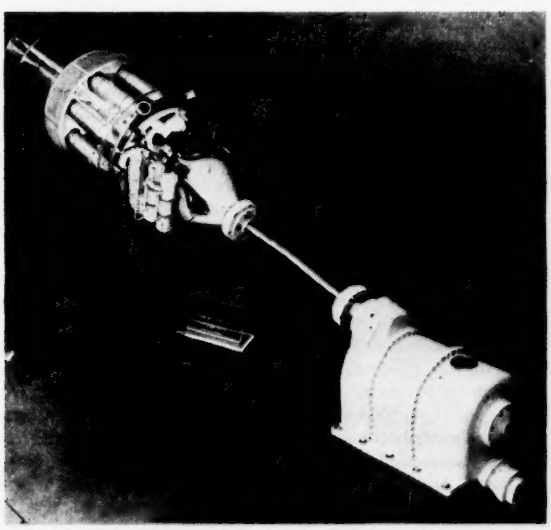


Fig. 5 Proteus 3500-hp marine gas-turbine engine. This type is being installed in British Naval high-speed craft, Brave Boats. (Courtesy Bristol Aero Engines Ltd.)

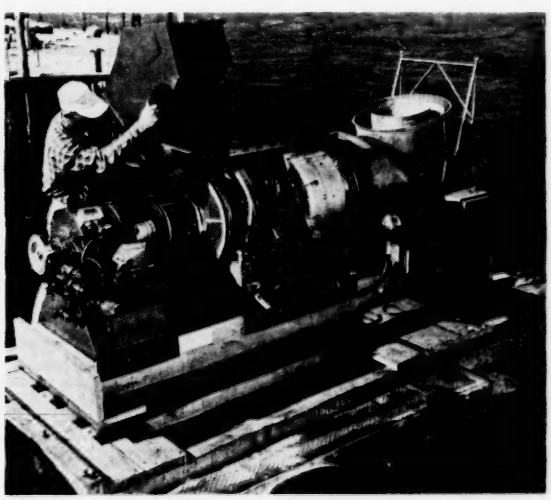


Fig. 6 T-53 gas-turbine marine power package for installation in a high-speed boat (Courtesy Lycoming Division, Avco)

time spent in port has been considerably less than 20 per cent" [19].

The Dart turboprop engine in Capital Airlines Viscount has shown an average bare-engine overhaul cost of $5000, made up of 23 per cent labor and 77 per cent materials. The average man-hours to accomplish this overhaul are 490. Time between overhauls is 1500 hr [20].

Fewer Spare Engines Needed. Experience with gas turbines, rated less than 2000 hp, has shown that the number of spare engines needed to support a ship or boat program is small. The general policy of one activity has been to procure 25 per cent spare gas turbines which is considerably less than the number required to support diesel prime movers [11]. Reliability of the Dart turboprop engine has permitted spares to be reduced by 14 [20].

Good Maneuverability. Tests conducted on the *John Sergeant* show the Liberty ship with the gas-turbine plant with CP propeller to have certain advantages over the *Thomas Nelson* (diesel ship) or the standard C2-class ship. For example during the crash-astern tests the following was recorded [21]:

	C2	*Thomas Nelson*	*John Sergeant*
Time to dead in water from 17 knots ahead, min:sec	3:25	3:36	2:52
Ahead reach, yards	1071	1103	757(Raydist) [21]

Availability. Operation of marine gas turbines during the past 5 yr shows good availability. Based on the first 20,000 hr of operation in naval service, the Boeing 502-6 had an availability of 96 per cent. This was from November, 1952, through December, 1954, and components have improved and availability increased since then [17]. Ten General Electric 6000-hp industrial units, recently completed a year of operation with an availability greater than 99 per cent [6].

The French merchant ships, *Cantenac* and *Merignac*, equipped with GS-34 free-piston gas-turbine-propulsion plants were immobilized for only 8 days in $2^1/_2$ yr and for 4 days in 2 yr of operation, respectively [22].

Overhaul Interval. With improvements in engine components resulting from continuing developments, overhaul intervals have increased more than 100 per cent on some engines. Overhaul period on one 160-hp marine gas turbine was set at 300 hr [17]. With an improved first-stage rotor, a 0.6-in. chord blade, this engine was tested for 1000 hr in the laboratory with only *routine* inspections and maintenance [23]. Later tests brought the total running time to 2500 hr without overhaul. The overhaul interval is now set at 750 hr, and later versions have 900-hr overhaul periods. Solar 500-hp engines, T-522 and T-520, have overhaul intervals of 1500 hr. A Brown-Boveri 5000-kw-capacity industrial-type engine installed in a power station at Dhahran has over 15,000-hr operation and has not been inspected [13].

Low Lubricating-Oil Consumption. Sea experience on the *Auris* has shown the marine-gas-turbine lubricating-oil consumption to be negligible. During four years of operation only one oil change was made and the oil had not deteriorated [19]. Lubricating-oil consumption for four U. S. Maritime ships with different type power plants is as follows:

Ship	Plant	Lube-oil consumption, gal/day (actual)
Benjamin Chew	Steam turbine	7
Thomas Nelson	Diesel	40
William Patterson	Free piston	80
John Sergeant	Gas turbine	3

Operation on Union Pacific locomotives gives a relative lubricating-oil consumption for different power plants on a per-1000-gross ton-mile basis [4]:

Plant	Relative lube-oil consumption
Steam	3.13
Diesel	2.98
Gas turbine	1.00

Initial Cost. Studies have shown that the installed cost of a gas-turbine propulsion plant in sizes usually found in merchant vessels should be less than steam-turbine plants [6]. In small gas turbines of less than 2000 hp, built along relatively lightweight designs, the initial costs have dropped sharply in the last several years.

New Uses. The versatility of the gas turbine is now more generally appreciated by potential users. It can provide a variety of services including: Electric power, shaft power, thrust, pneumatic power, air-turbine drive, provide cooling through air-cycle cooling turbines, start other engines, and supply heat. Marine installations of gas turbines now include: Main propulsion, emergency-generator drive, fog generator, deicers, air supply, fire pump, and electric power for tanker cargo-pump motor. Some of these applications are newcomers to ships, but they have already demonstrated their ability to do a better job than the prime mover they replaced.

Engines in Use or Building for Installation

Gas-turbine and free-piston plants that have been installed or building for marine service during the past several years exceed 313,000 shp, Table 1, descriptions of representative installations will be given although some have been described elsewhere in considerable detail.

Merchant-Propulsion Gas Turbines. Gas turbines for propulsion of merchant vessels exceed 25,000 hp. Numerous plants are shown in Table 2. A number of propulsion plants will be described briefly:

1 *Auris.* The new plant is a British Thomson-Houston Company turbine, rated at 5500 bhp. It is equipped with hydraulic mechanical drive. This propulsion unit is of the open-cycle, two-shaft, straight-compounded, intercooled, regenerative type, Fig. 7. Shaft power is taken from the compressor end of the low-pressure shaft. The high-pressure and low-pressure shafts are line-in-line and are raked several degrees toward the stern. Air enters an inlet silencer and then a 12-stage axial-flow compressor. This compressor is driven by the l-p 8-stage turbine. From the l-p compressor air is discharged to an intercooler and then to the h-p compressor. This compressor has 16 stages and is driven by a 5-stage turbine. Air then enters the contraflow regenerator and flows into the single combustion chamber. The combustion chamber is of reverse-flow type and is hung under the h-p turbine. From the h-p turbine gas flows to the l-p turbine through six flexible tubular ducts set around the shaft center line. After leaving the l-p turbine, the gases flow through the tubes of the heat exchanger. A 5000-lb-per-hr 50-psi waste-heat boiler receives the exhaust gases after they leave the heat exchanger. Gases are then discharged to the stack. The turbine-inlet temperature is 1200 F with an inlet of 75 F. The over-all thermal efficiency is 28 per cent, excluding the waste-heat boiler.

The plant is started by a small steam turbine driving the h-p shaft through a reduction gear. Power is supplied to the propeller through the l-p shaft. While at sea, auxiliary electric power will be supplied by a 200-kw steam turbine-generator set taking steam from the waste-heat boiler. A 550-kw generator can be connected to the main reduction gear for pumping operations while at sea or in port. An emergency-propulsion steam turbine rated at 450 shp is provided, which can be connected to the propeller through a separate reduction gear. Steam is supplied to this turbine from a small oil-fired boiler [24].

2 *John Sergeant.* The propulsion plant of the *John Sergeant*

Table 2 Gas turbines in merchant propulsion application

Country	Manufacturer of engine	Model	Power/ plant, hp[a]	Plants/ ship	Ship	No. of ships	Status	Astern power
England	British Thomson-Houston..	...	1200	1	*Auris*	1	Removed after 19831 hr	Electric drive
England	British Thomson-Houston..	...	5500	1	*Auris*	1	Installed	Hydraulic mechanical
England	Blackburn.....	Turmo 600	350	2	Launch	1	Building	Reverse gear
France	Rateau-A.C.B.....	...	3300	1	Liberty	1	Building	CPP
Italy	Metro-Vickers Electric Co., Ltd.....	...	5000	1	...	1	Building	...
U. S.	Boeing Airplane Co.....	502-10C	220	2	Personnel boat	1	Installed	Reverse gear
U. S.	Solar Aircraft Co.....	T-522	500	2	Personnel boat	1	Building[b]	Reverse gear
U. S.	Solar Aircraft Co.....	T-522	500	2	Personnel boat	1	Building[c]	Reverse gear
U. S.	General Electric Co.....	...	6600	1	*John Sergeant*	1	Installed	CPP

[a] Plant includes a turbine or turbines supplying power to a single shaft.
[b] Building by Sewart Seacraft.
[c] Building by Higgins.

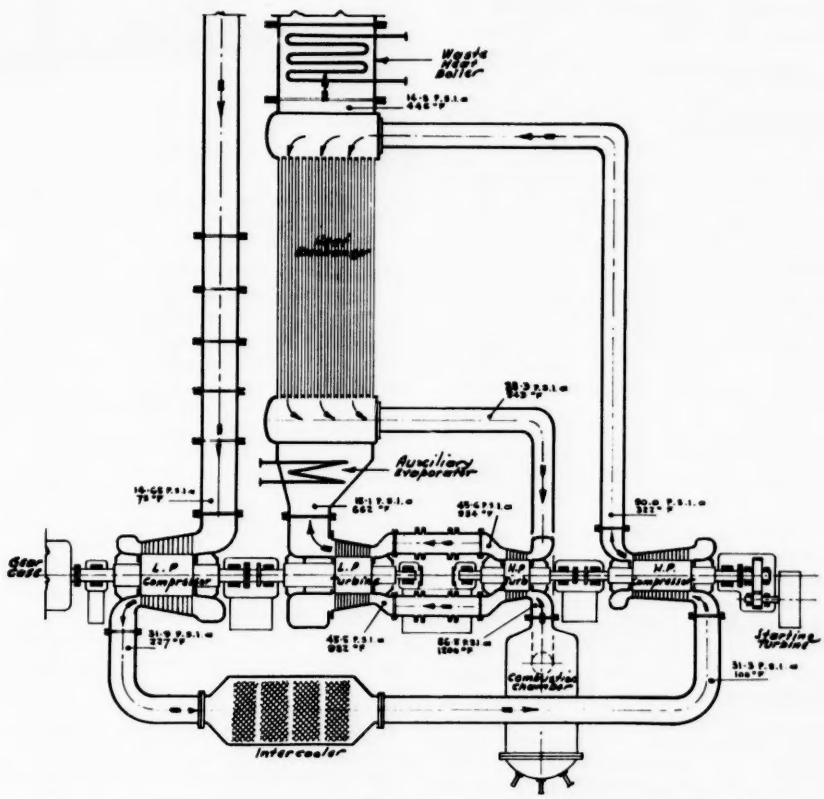


Fig. 7 Diagram of the British Thomson-Houston 5500-hp gas-turbine engine in the *Auris* [54]

Fig. 8 The 6600-hp gas turbine installed in *John Sergeant* (Courtesy General Electric Company)

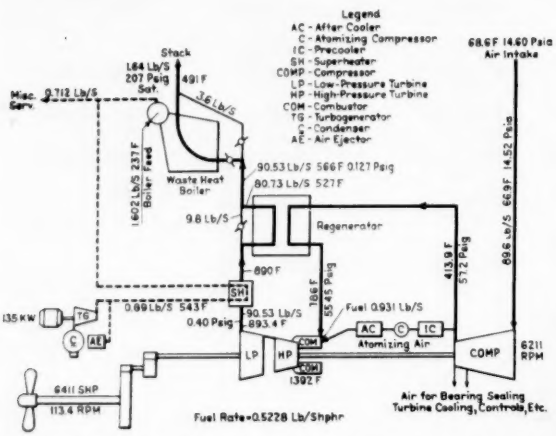


Fig. 9 The gas-turbine cycle on the *John Sergeant* (Courtesy The Society of Naval Architects and Marine Engineers)

includes an open-cycle 2-shaft regenerative gas turbine, Fig. 8. This turbine has an axial-flow 14-stage compressor. It is driven by the h-p turbine. A separate l-p free-power turbine, in the same casing with the h-p turbine, supplies power to the controllable-pitch propeller through a reduction gear. Air flow through the machine is as follows: Air enters the axial-flow compressor, is compressed to about 57 psig; discharged to the regenerator where it is heated by exhaust from the l-p turbine; then to the combustors and to the h-p turbine; exhaust from the h-p turbine enters the l-p unit; flows through the superheater and into the extended surface "pin-fin-type" regenerator which has an effectiveness of 80 per cent, where it gives up heat to the compressed air prior to combustion; then through the superheater and waste-heat boiler and out to the stack. The cycle is shown in Fig. 9.

The turbine, a General Electric machine, Fig. 8, was designed for a normal power rating of 6000 shp. It has a maximum continuous rating of 6600 shp at an inlet-air temperature of 80 F. Turbine-inlet temperature at 6000 shp is 1450 F. The gas generator has a full-load rated speed of 6900 rpm. The l-p power turbine has an inlet design temperature of 950 F and a maximum load speed of 5517 rpm. At this speed the propeller turns at 112.5 rpm. Variable-area nozzles are installed in the power turbine for easier control and to improve efficiency. Starting power is provided by a 290-hp steam turbine. The propulsion plant, gas turbine, and controllable-pitch propeller are controlled in the engine room by either a co-ordinated single lever or independently by separate levers. Single-lever control can only be exercised from the bridge.

Astern power is obtained through the controllable-pitch propeller. An emergency power plant consists of a 750-hp steam turbine that could be connected to the main reduction gear if needed [25]. On-board fuel-oil treatment is necessary to minimize corrosion and deposition caused by sodium, calcium, and vanadium which are present in the residual fuels [26].

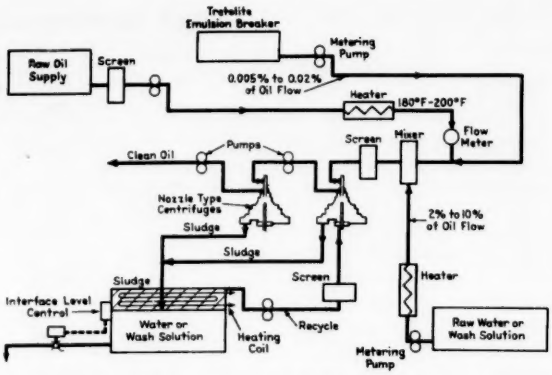


Fig. 10 Fuel-treating system on the *John Sergeant* (Courtesy The Society of Naval Architects and Marine Engineers)

Fig. 11 Personnel boat, Creole Petroleum Company, powered by two Boeing 220-hp gas turbines (Courtesy Boeing Airplane Company)

The system on the *John Sergeant*, Fig. 10, removes sodium and calcium by intimately mixing about 5 per cent fresh wash-water with heated fuel. The wash-water with its dissolved sodium and calcium is removed by centrifuging. A demulsifying agent is used to aid the separation process. A magnesium-sulfate-water solution is mixed into the "cleaned" fuel just ahead of the turbine fuel pump at a rate of 2 to 3 per cent of fuel to obtain the required magnesium-to-vanadium ratio [25].

Personnel Boats. A number of commercial small craft have been equipped with gas turbines for propulsion. Two Boeing engines, 502-10C, rated at 220 hp, have been installed in a small boat for the Creole Petroleum Company, Fig. 11. This is a 2-shaft installation. Astern power is obtained through a commercial-type reverse gear. A Blackburn and General Aircraft engine, Turmo 600, rated at 350 hp, is being installed in a launch. This installation will be equipped with a reverse gear for astern power. Two boats, Figs. 12 and 13, now under construction, will have two 500-hp Solar gas turbines for propulsion. These boats will use commercial-type reverse gears for astern power. Characteristics of the boats are given in Table 3.

Merchant-Propulsion Free-Piston Gas Turbine. This prime mover now totals about 69,000 hp, Table 1, either in service or building. Plants are listed in Table 4. Some existing plants are briefly described in the following paragraphs:

1 *Cantenac and Merignac.* These two French coasters, 920-ton deadweight, length over-all 208 ft, are the first ships propelled by the free-piston gas turbine. They have been in service since February and August, 1954. The propulsion plant consists primarily of two GS-34 gasifiers and two gas turbines supplying power to a propeller through a combining reduction gear. Data on the GS-34 gasifier is given in Table 5. The GS-34 gasifiers

were built by Société Industrielle Générale de Mécanique Appliquée. The turbines were constructed by Alsthom. The gas enters the turbines at about 50 psi and a temperature in the range 845–935 F. The turbines operate at 9000 rpm and develop 800 hp. The turbine has three rows of ahead blading and one row of astern power. The propeller turns at 220 rpm. The power output has been limited to 1200 hp because of the hull form [27].

2 *William Patterson.* This is one of four Liberty ships converted by the U. S. Maritime Administration as a part of its ship and engine-improvement program. The propulsion plant includes: Six GS-34 gas generators, two 3000-shp gas turbines, each with astern elements and one main reduction gear. For economy of conversion many of the original auxiliaries were retained in the ship. Replacements were limited only to those required for the main propulsion plant. Characteristics of the gasifiers are shown in Table 5. Turbine data are given in Table 6. Inlet gas temperature to the turbine is 875 F. The turbines have six ahead and two astern stages. During ahead operation the astern elements are masked to reduce windage losses. The plant is controlled by one handwheel on the control board for all powers ahead and astern. The turbine rated speed is 5500 rpm [28].

3 *Ormara.* A conversion of this 5400-ton ship, built in 1947, from steam to free-piston-gas-turbine propulsion is being made. Four GS-34 free-piston gasifiers will be installed. Gas from these units will enter a gas turbine that is geared to the propeller. The turbine is of British Thomson-Houston design and manufacture. It is rated at 4000 shp, 6800 rpm. There are six stages reaction in the ahead portion. A single-stage impulse turbine is flexibly coupled to the forward end of the ahead turbine for astern power. This turbine is capable of delivering astern power equal to approximately 60 per cent ahead power. The turbine is coupled to a double-reduction gear with an output speed of 110 rpm [29].

4 *French Merchant Ships.* Two new ships are being built by la Marine Nationale Française with free-piston-gas-turbine-propulsion plants. These ships will have an installed power of 16,000 shp and will use GS-34 gasifiers [30].

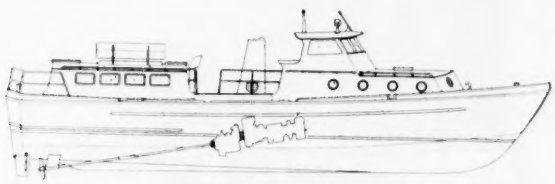


Fig. 12 Personnel boat being built by Sewart Seacraft, powered by two Solar T-522 gas turbines; total shp 1000, speed greater than 30 knots (Courtesy Solar Aircraft Company)

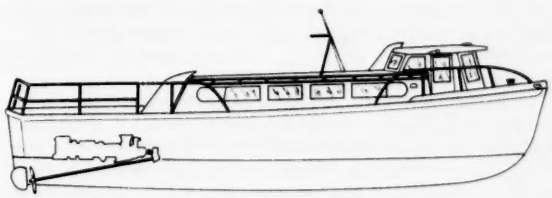


Fig. 13 Personnel boat being built by Higgins. Powered by two Solar T-522 gas turbines; total shp 1000, speed greater than 40 knots. (Courtesy Solar Aircraft Company)

Table 3 Commercial personnel boats under construction

Boat builder	Sewart Seacraft	Higgins
Engines	Solar T-522	Solar T-522
Total shp	1000	1000
LOA	55 ft	52 ft 10 in.
Beam	15 ft 1 in.	15 ft 10 in.
Speed, knots	30 plus	40 plus
Personnel capacity	28	30
Fuel capacity, gallons	700	700
Number shafts	2	2

Table 5 Design data on free-piston gasifiers

Item	GM-14, or GS-34	CS-75
Engine cylinder bore, in.	13.4	7.5
Compressor cylinder bore, in.	35.4	20.75
Over-all length, ft	14	7.5
Stroke, cmr[a]	2 × 18 in.	2 × 9.5 in.
Number cycles per min, cmr	600	1000
Mean piston speed, fpm	1800	1580
Gas temperature, cmr	840 F	470 C
Gas horsepower, cmr	1230	420
Weight, tons	7.5	2.25
Gas flow, lb per sec	8.5	...

[a] cmr = continuous maximum rating.

Table 4 Free-piston gas turbines in marine installations

Country	Manufacturer	Gasifier model	Power/ plant, shp[a]	Plants/ ship	Service[b]	Ship	No. of ships	Status	Gasifiers-turbines/ plant[c]	Astern power
England	Free Piston Engine Co.	GS-34	2500	1	MP	Ore carrier	1	Building	G3/T1	Turbine
England	A. Stephens Sons	GS-34	4000	1	MP	*Ormara*	1	Building	G4/T1	Turbine
England	...	GS-34	4000	..	MP	...	1	Building	...	...
England	...	GS-34	2000	1	MP	...	1	Building	...	...
England	Free Piston Engine Co.	GS-34	7000	1	MP	Crest Ship Co.	1	Building	G-/T2	...
England	Smiths Dock	GS-34	...	1	MP	Bolton S.S. Co.	1	Building	G4/T-	...
England	Free Piston Engine Co.	CS-75	200 kw	1	MA	...	..	Building	G1/T1	...
France	SIGMA	GS-34	900	2	NP	Minesweep	21	Installed	G2/T1	Turbine
France	...	GS-34	8000	2	MP	...	2	Building	G2/T4	CPP
France	...	GS-34	2000	1	MP	*Goodwood*	1	Building	G2/T1	...
France	SIGMA	GS-34	2000	1	MP	*Merignac Cantenac*	2	Installed	G2/T2	Turbine
France	SIGMA	GS-34	600 kw	1	MA	*Bethsabee*	1	Installed	G1/T1	...
Germany	Demag Modag	GS-34	1400	1	MP	*Sagitta*	3	1 in operation, 2 building	...	Turbine
U. S.	SIGMA-GM	GM-14	6000	1	MP	*W. Patterson*	1	Installed	G6/T2	Turbine

[a] Plant includes gasifiers and turbines supplying power to a single load.
[b] M = merchant, N = naval, P = propulsion, A = generator.
[c] G = gasifiers, T = turbines.

Auxiliary. Merchant use of auxiliary gas turbines now totals about 6000 hp in generator drives and pumps, Tables 1 and 7.

1 *350-kw Generator.* This gas-turbine-driven generator set was designed and manufactured by the W. H. Allen and Sons Company, Ltd. Data on the gas turbine Type D, Fig. 14, is presented in Tables 7 and 8. This is a single-shaft machine. The compressor is single-stage-centrifugal with a pressure ratio of 2.7:1. The speed of the turbine is 15,000 rpm. The generator is driven at 1500 rpm through a reduction gear coupled to the compressor end of the machine. The set has automatic starting and is brought on the line in the event of failure of the ship's power supply. The set reaches full speed in 35 sec. Starting is accomplished by an air motor supplied from air bottles under 350-psi pressure.

2 *Fire Pump.* A 500-gpm fire pump driven by a Solar T-45 gas turbine, Fig. 15, was installed on the *Thomas Nelson* by the Maritime Administration. This pump, normally a portable unit, was installed on a fixed foundation in the shaft alley. The unit is handcranked, burns diesel fuel, weighs about 180 lb and delivers 500 gpm at 100 psi. Exhaust gases are ducted to the upper deck and discharge to atmosphere.

3 *300-kw Generator.* A 300-kw-capacity gas-turbine-driven generator set, Fig. 16, is being installed on the *M. V. Weybridge.* The set is driven by a Ruston and Hornsby T-E gas turbine. The engine characteristics are: Weight 4400 lb, dimensions 90 × 48 × 69 in., turbine-inlet 1470 F, pressure ratio 3.5:1. This unit has provisions to operate in conjunction with a waste-heat boiler. The specific fuel consumption is 0.98 lb per hp-hr. A second set is to be installed in another vessel.

Merchant-Auxiliary Free Piston. Merchant applications of the free-piston gas turbine for auxiliary drives are increasing.

1 *600-kw Generator Set.* A GS-34 free-piston gasifier and a Rateau 6-stage gas turbine are used to drive a 600-kw generator

Table 6 Characteristics of the gas turbines in the free-piston-powered ship *William Patterson* [28]

Number of ahead stages	6
Mean diameter, last-stage	25 in.
Blade height, last-stage	$4^3/_4$ in.
Mean diameter, first-stage	$22^3/_8$ in.
Blade height, first-stage	$2^1/_8$ in.
Rotor mean diameter	$10^3/_8$ in.
Number of astern stages	2
Mean diameter, last-stage	$16^5/_8$ in.
Blade height, last-stage	$2^7/_8$ in.
Rotor shaft length between bearing centers	4 ft $10^3/_4$ in.
Turbine casing, over-all length	7 ft 1 in.
height	4 ft $6^3/_8$ in.
width	4 ft 6 in.

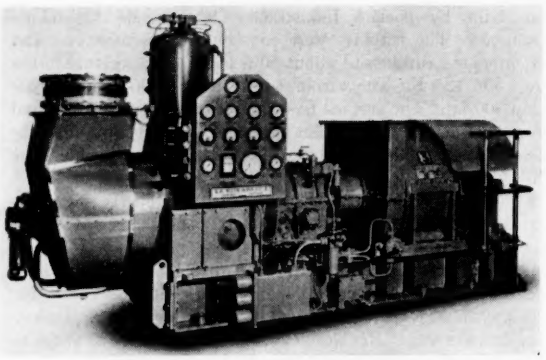


Fig. 14 Gas-turbine-driven 350-kv emergency-generator set. Units have been installed in five commercial ships. (Courtesy W. H. Allen Sons and Company, Ltd.)

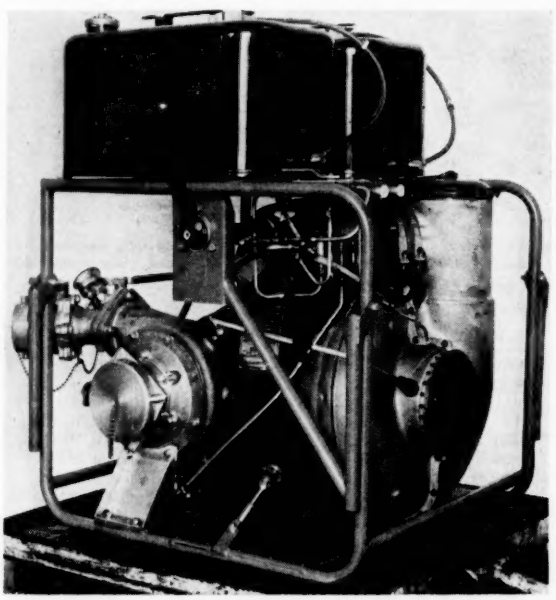


Fig. 15 Portable fire pump 500-gpm capacity driven by a Solar T-45 gas turbine (Courtesy Solar Aircraft Company)

Table 7 Gas turbines—merchant auxiliary applications

Country	Manufacturer of engine	Model	Power/ plant	Plants/ ship	Ship	No. of ships	Status	Service
England	W. H. Allen Sons & Co., Ltd.	Type D	350 kw	1	*Dolius* class	5	Installed	Emergency generator
England	John Brown	...	200 kw	1	New Zealand Ship Co.	1	Building	Generator
England	Ruston & Hornsby, Ltd.	T-E	300 kw	1	M. V. Weybridge	2	Building	Generator
England	Ruston & Hornsby, Ltd.	T-A	1300 hp	1	...	1	Installed	Generator
Holland	W. H. Allen Sons & Co., Ltd.	Type D	350 kw	1	*SS Rotterdam*	1	Building	Generator
U. S.	Solar Aircraft Co.	T-45	46 hp	1	*Thomas Nelson*	1	Installed	Fire pump
U. S.	...	...	600 kw	1	*Marad*	1	Under procurement	Generator
Japan	Boeing Airplane Co.	...	220 hp	1	...	1	Delivered	Starting turbine

Table 8 Gas turbines available for marine applications [42]

Manufacturer	Manufacturer's model number	Horsepower continuous rating	Type turbine	Pressure ratio	Fuel consumption, lb per shp-hr	Engine dry weight, lb	Hours between overhaul, C.R.[a]
AiResearch	GTCE 85-2	Air bleed 109 lb per min at 101″ Hg	Single-shaft	3.47	(226 lb per hr)	550	500
	GTC 85-24	Air bleed 109 lb per min at 101″ Hg	Single-shaft	3.47	(218 lb per hr)	190	500
	GTP 250-2	262	Single-shaft	3.5	1.13	316	1000
	GTCP 85-20	Air bleed 103 lb per min + 170 shp at 106″ Hg	Single-shaft	3.6	1.54	235	...
	GTP 70-10	89	Single-shaft	3.3	1.38	190	1000
W. H. Allen	Type A	1340	Free-power turbine	...	0.88	19500	10000
	Type B1	130	Single-shaft	2.5	1.71	650	1000
	Type C	670	Single-shaft	9.69	0.82	8100	5000
	Type D	400	Single-shaft	2.7	1.42	2250	1000
Blackburn General	Palouste 500	...	Single-shaft	3.92	(274 lb/hr)	204	500
	Artouste 600	380	Single-shaft	4.12	1.00	206	500
	Turmo 600	370	Free-power turbine	4.12	1.09	216	500
Boeing	502-6	160	Free-power turbine	3	1.4	315[b]	500
	502-8C	160	Free-power turbine	3	1.4	298[b]	500
	502-10C	220	Free-power turbine	4	1.06	325[b]	500
	502-11B	Air bleed, 192 air hp	Free-power turbine	4	1.20 lb per air hp-hr	350	...
	520-1	300	Free-power turbine	6.2	0.80	425	2000
David Budworth		50	Single-shaft	2.8		90	1000
Clark Bros.	302	9300	Free-power turbine	5	0.65	140000	100000
	TA	1150	Free-power turbine	4	0.91	12200	100000
Continental Aviation	141	2.1 lb per sec bleed air	Single-shaft	3.5	280 lb per hr	210	200
Curtiss-Wright	J65W16 Marine	8250	Free-power turbine	5.88	0.68	6750	2000
English Electric Co.	EM27P	2500	Free-power turbine	4.9	0.59 reg.	30100	10000
	EM27P	2680	Free-power turbine	4.9	0.79	20200	10000
General Electric	T-58	800	Free-power turbine		0.69	267	...
		6700	Free-power turbine		6.98×10^6 Btu/hr		...
	7LE70	7000	Single-shaft	6.0		75000	...
		7150	Free-power turbine		98.9×10^6 Btu/hr		...
	7LA75	7500	Single-shaft	6.0		75000	...
		7600	Free-power turbine		73.2×10^6 Btu/hr		...
	SC2S	8000 (gas only)	Free-power turbine		105.5×10^6 Btu/hr		...
	7LA134	13400	Single-shaft			125000	...
	7LA225	22500	Single-shaft			251000	...
	7LA304	30400	Single-shaft			398000	...
Lycoming	LTC4A-1	1112	Free-power turbine		0.800	695	...
	LTC4B-1	1165	Free-power turbine		0.758	600	...
	LTC1B-1	685	Free-power turbine	6.0	0.75	480	...
	LTC1-F-1	734	Free-power turbine		0.739	460	...
Metro-Vick	G-2	3500	Free-power turbine		0.88	6750	300
	G-4	4500	Free-power turbine	6.3	0.69	8650	500
Ruston & Hornsby	TE	420	Single-shaft	3.5	1.01	4200[b]	8000
Solar	T-41M	45	Single-shaft	2.35	2.0	45	500
	T-45M8	45	Single-shaft	2.56	2.0	45	600
	T-45M	0.625 lb per sec at 18 psig (air bleed)	Single-shaft	2.25	100 lb per hr	45	500
	T-300	150 lb per min at 60 psig (air bleed)	Single-shaft	4.1	350 lb per hr	725[b]	500
	T-520	450	Single-shaft	4.6	0.96	550	1000
	T-522	450	Free-power turbine	4.6	0.96	580	1000
	T-1000	1000	Free-power turbine	6.5	0.65	725	1000
Utica Bend	100ST	3500	Free-power turbine	7.25	0.65	6800	5000
	200ST	2580	Free-power turbine	6.4	0.67	2450	2000
	300ST	50	Single-shaft	2.9	1.8	135	1000
Westinghouse	W-31	3250	Single-shaft	5	0.72	27600	9000
	W-52-R	5000	Free-power turbine	4.07	0.58 reg.	213700	9000
	W-52	6000	Free-power turbine	4.07	0.82	60000	9000
	W-81-R	7500	Single-shaft	6	0.55 reg.	318000	9000
	W-81	8000	Single-shaft	6	0.68	63700	9000
	W-121	12000	Single-shaft	6	0.65	173000	9000
	W-201	22500	Single-shaft	6	0.72	224500	9000
	J40WE22AG	7500	Free-power turbine	4.45	0.86	8075	500

[a] C.R. = Continuous Rating. [b] Including reduction gear.

set on the French tanker *Bethsabee*. This set was installed in the ship in June, 1952.

2 *200-kw Generator Set.* A number of generator sets, Table 4, driven by CS-75 free-piston gas generator and a gas turbine are being produced for the New Zealand Shipping Company, Ltd., and the Port Line, Ltd. The turbine is a 3-stage axial-flow unit built by David Budworth, Ltd. The turbine develops 350 hp at 29,000 rpm. Rotor blades are forged of heat-resisting high-tensile-strength steel. Stator blades are precision cast. Inlet guide vanes are adjustable. Total weight, dry, is about 14,000 lb. Specific fuel consumption is about 0.408 lb per shp-hr [31].

Naval Propulsion Gas Turbines. Excellent experience with a number of propulsion installations of gas turbines made by The British Admiralty has, without doubt, led to wide application of gas turbines in the marine field. U. S. Navy work has also been instrumental in furthering the use of this prime mover. The total power installed, or building, for naval propulsion is now about 124,000 hp, see Tables 1 and 9.

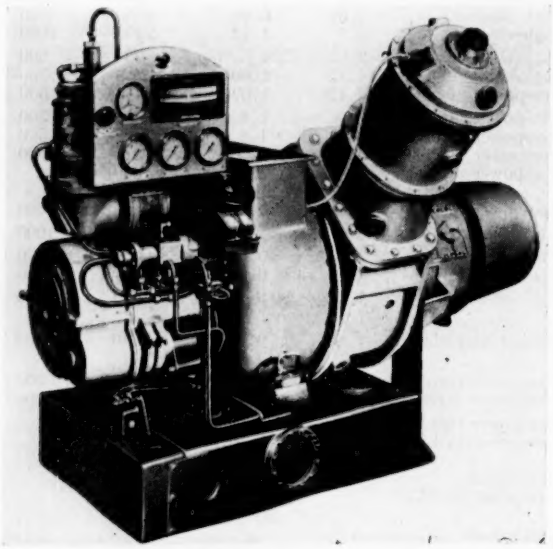


Fig. 16 300-kw generator set driven by gas turbines, Model TE, installed in two commercial ships (Courtesy Ruston & Hornsby, Ltd.)

HMS Grey Goose. This gunboat has a length over-all of 146 ft, a 20-ft beam, and a displacement of 205 tons [32]. The British Admiralty installed two Rolls-Royce gas turbines, RM-60, in the *Grey Goose* for propulsion, Fig. 17. The RM-60 comprises an 11-stage axial-flow l-p compressor, discharging through twin intercoolers in parallel. Air passes to an h-p centrifugal compressor, two stages with intercooling. The h-p compressor discharge enters a heat exchanger. Air then goes to the combustion chambers, to the h-p turbine, the intermediate power turbine which drives the propeller, and then to the l-p turbine. From the l-p turbine, the exhaust gases flow through the heat exchanger and then to atmosphere. Astern power is provided through Rotol 3-bladed controllable-pitch propellers. Complete plant description has been presented earlier [10].

Minesweeping Launch. The U. S. Navy's 36-ft minesweeping launch is propelled by a Boeing 502-8C gas turbine rated at 160 hp. Four boats of this class were built initially. Three were equipped with hydraulically actuated reverse gear, and the fourth has a controllable-pitch propeller [33]. Twenty-six new boats being built, Fig. 18, are equipped with Boeing 502-10C gas turbines, rated at 220 hp, for propulsion. Astern power is supplied by a reverse reduction gear.

HMS Brave Borderer. This boat, Fig. 19, is the first of the new Brave-class fast patrol boats. It was launched on January 7, 1958. The extreme length of the boat is 98 ft 10 in., beam is 25 ft $5^1/_2$ in. This craft is powered by three Proteus marine gas turbines, Fig. 5, manufactured by Bristol Aero Engines, Ltd. Primary reduction gear and the reverse reduction gear are manufactured by W. H. Allen and Sons, Ltd. The marine rating of the Proteus is 3500 shp max for 30 min and 2800 shp continuous. The specific weight of the propulsion plant is 1.6 lb per hp [34]. The engine-and-reduction-gear weight is 2900 lb. This is a specific of 0.83 lb per hp. The engine can be started from a cold

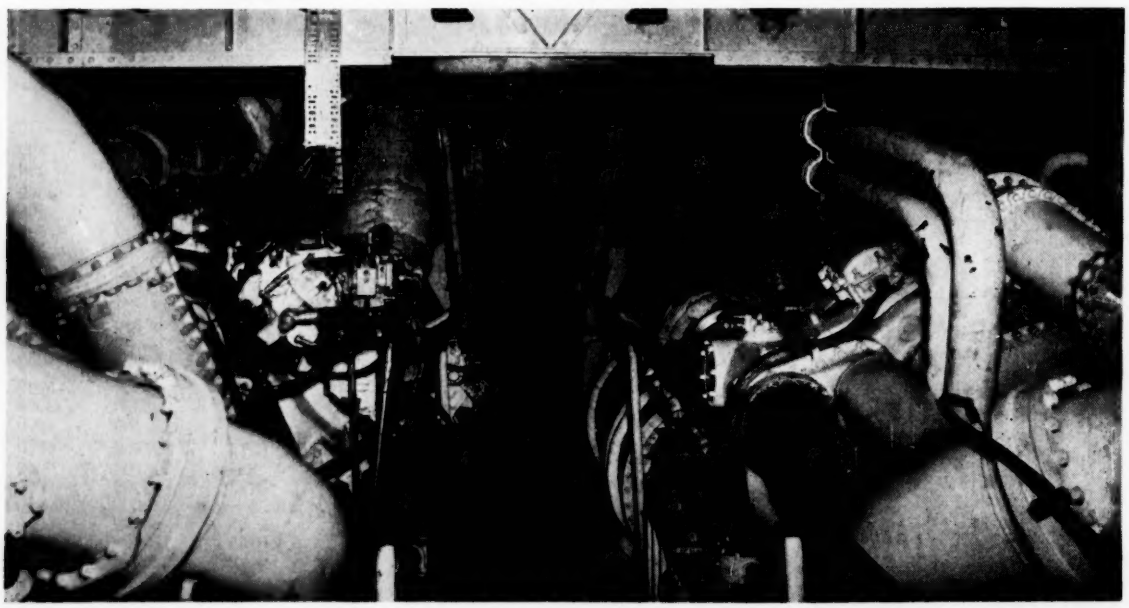


Fig. 17 Gas-turbine propulsion engines, Rolls-Royce 5400-hp RM-60, installed in *HMS Grey Goose* (Courtesy British Admiralty)

condition in 30 sec. It can be brought from idle to maximum power in 30 sec [9]. Electrical power for the craft is supplied by 40-kw generators driven by Rover gas turbines. It is interesting to note that there is no diesel machinery on this craft.

PT-812. The U. S. Navy has replaced the reciprocating engines in this vessel. There are two 4000-hp Metropolitan Vickers G-2 gas turbines—one each on the two outboard shafts, and there are two 600-hp diesels—one each on the two inboard shafts. There are no clutches in the outboard shafts. Astern power is provided by controllable-pitch propellers on the two diesel-driven shafts. In this plant the diesels provide needed power up to 18 knots. The gas turbines are used for speeds greater than the 18-knot cruise condition. Gas turbines can be controlled from either the bridge or engine room. Diesel fuel is used for both types of propulsion engines. A gas-turbine-driven Solar T-45 portable fire pump is also installed.

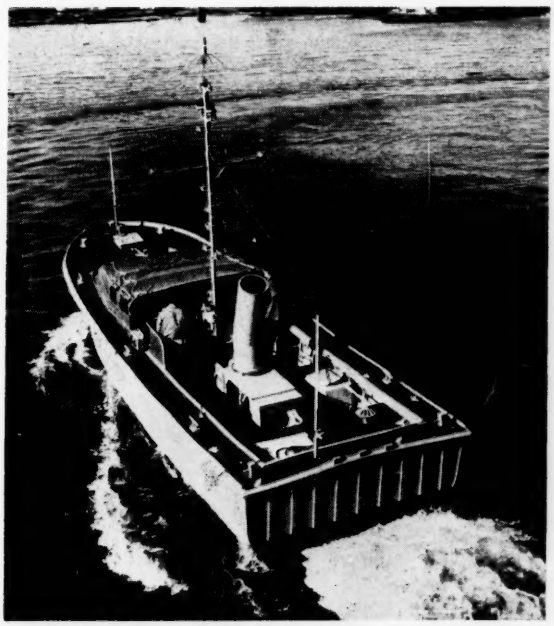


Fig. 18 U. S. Navy MSL powered by Boeing 502-10C 220-hp gas turbines (Courtesy Boeing Airplane Company)

Landing Craft Personnel (L). The U. S. Navy recently installed a Solar 500-hp gas turbine in a new 40-ft-long LCPL, Fig. 20. The decision to make this prototype installation was based on the excellent operating experience gained with a similar engine installed in a 40-ft personnel boat [35]. During the operational

Fig. 19 *HMS Brave Borderer* powered by three Proteus marine engines (Courtesy British Admiralty)

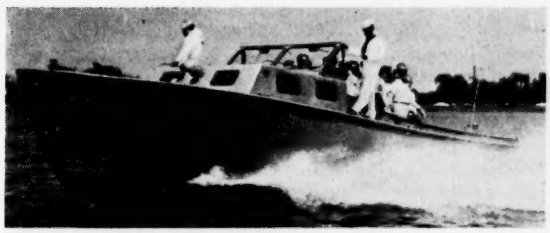


Fig. 20 LCPL powered with a 500-hp Solar gas turbine (Courtesy U. S. Navy)

Table 9 Gas turbines in naval propulsion applications

Country	Manufacturer of engine	Model	Power/plant, hp[a]	Plants/ship	Ship	No. of ships	Status	Astern power
England	Rolls-Royce Ltd.	RM-60	5400	2	*HMS Grey Goose*	1	Installed	CPP
England	Metro-Vickers Electric Co., Ltd.	G2/1	4500	2	MTB/MTG	2	Installed	Diesel
England	...	...	Boost unit	..	Frigate	..	Building	Steam turbine
England	...	...	Boost unit	..	Destroyer	..	Building	Steam turbine
England	Bristol Aero Engines Co., Ltd.	Marine Proteus	3800	3	Fast Patrol Boat	..	1 boat complete	Reverse gear
U. S.	Boeing Airplane Co.	502-10C	220	1	LCVP	4	Installed	Reverse gear
U. S.	Pratt & Whitney Aircraft	T-34	5000	4	Liberty	1	Installed	Air properties
U. S.	Metro-Vickers Electric Co., Ltd.	G2/2	4000	2	PT	1	Installed	CPP
U. S.	Solar Aircraft Co.	T-522	500	1	40-ft PB	1	Installed	Reverse gear
U. S.	Solar Aircraft Co.	T-522	500	1	LCPL	2	1 building 1 installed	Reverse gear
U. S.	...	...	500	1	LCPL	7	Procuring	Reverse gear
U. S.	Boeing Airplane Co.	502-10C	220	1	MSL	26	Installed	Reverse gear
U. S.	Boeing Airplane Co.	502-6	160	1	LCVP	1	Installed	Reverse gear
U. S.	Boeing Airplane Co.	502-8C	160	1	MSL	4	Installed	3 Reverse gears 1 CPP
U. S.	Lycoming Division of Avco	T-53	685	1	Hydrofoil	1	Building	Reverse gear
Japan	Escher Wyss	JN-10	10000	2	...	..	Building	CPP
Japan	Mitsubishi	...	...	1	...	1	Building	...
U. S.	Boeing Airplane Co.	502-10C	220	1	PB	1	Building	Reverse gear
Germany	Brown-Boveri	...	...	..	Escort	6	Building	...

[a] A plant includes a turbine or turbines supplying power to a propeller, generator, or other load-absorbing means.

tests, the 40-ft boat traveled some 16,000 miles at speeds in excess of 25 knots. Astern power is supplied through a mechanical reverse reduction gear. The entire propulsion-plant weight is 2360 lb, including turbine, gears, and auxiliaries. This is a specific weight of 4.7 lb per shp. The boat has proved to be most satisfactory in operational evaluations. Boat speeds in excess of 23 knots have been obtained. LCPL's with diesel and gas-turbine propulsion plants are compared in Table 10. It is of interest to note that the gas-turbine boat is 5.8 knots faster. Also the total boat weight is 200 lb less for the gas-turbine-powered craft. Both boats have the same cruising range. As a result of the experience with this LCPL, eight new LCPL's, 36 ft in length, are being procured with 500-hp gas turbines for propulsion.

10,000-shp Prototype Closed-Cycle Plant. A closed-cycle propulsion plant, utilizing air as the working fluid, is under construction by Escher-Wyss. The plant is designed to use diesel fuel, and tests will also be conducted on residual fuel. Pressure ratio is 3.8:1. The compressor-inlet is 240 psia. The h-p turbine has four stages and an expansion-pressure ratio of 2:1 at full load. The power turbine has three stages and an expansion ratio of 1.6:1. The power-turbine speed is 7700 rpm with an overdrive of the propeller of 9300 rpm. A double-reduction gear connects the power turbine to a controllable-pitch propeller at 350 rpm. Maximum plant efficiency is realized under cruising conditions, 10 to 30 per cent of full power. Expected specific fuel consumption is 0.5 lb per shp-hr at 10 per cent power and 0.73 lb per shp-hr at full power [36]. Specific weight of the plant (including turbines, compressors, air heaters, auxiliaries, and reduction gear) is about 32 lb per shp.

Naval Free-Piston Propulsion. The French Navy, in 1953–1955, installed free-piston gas-turbine propulsion plants in 21 minesweepers. These ships are 2-shaft and each shaft is driven by a free-piston gasifier combined with a gas turbine. The gasifiers are GS-34 units. Turbines are by Rateau. The power per shaft is 900. Astern power is supplied by astern stages in the turbine. These installations are the only publicized naval propulsion plants using free-piston gasifiers combined with turbines.

Table 10 Comparison of gas-turbine and diesel-propelled 40-ft LCPL

	Diesel	Gas turbine
Full speed, knots	17.30	23.14
Towing LCM-6, knots	11.5	18.5
Cruising radius, miles	140	140
Horsepower	300	500
Specific fuel consumption, lb per shp-hr	0.43	1.04
Weight, lb		
Boat, total	19700	19500
Machinery	3115	2360
Fuel	1120	2100
Machinery plus fuel	4235	4460

Naval Auxiliary Gas Turbines. The use of gas turbines for auxiliary applications, Tables 1 and 11, in naval service is increasing at a rapid rate. The present applications total about 51,000 hp. These include air-supply units, fire pumps, fog generators, emergency, and stand-by generator sets [37].

1 *500-kw Generator.* The British Admiralty has installed a gas-turbine-driven 500-kw generator in the *HMS Llandaff*, Fig. 21. This is a base-load set. The Type C turbine, Tables 8 and 11, was designed and constructed by W. H. Allen Sons and Company, Ltd. It is a compound engine with intercooling. Air enters the l-p axial-flow compressor and is compressed to a 2.7:1 ratio. Then to the intercooler and to the h-p compressor where it is compressed to a pressure ratio of 3.8:1. Air enters the single combustion chamber at 131.9 psia. Combustion gases flow through the h-p turbine to the l-p turbine and then to atmosphere.

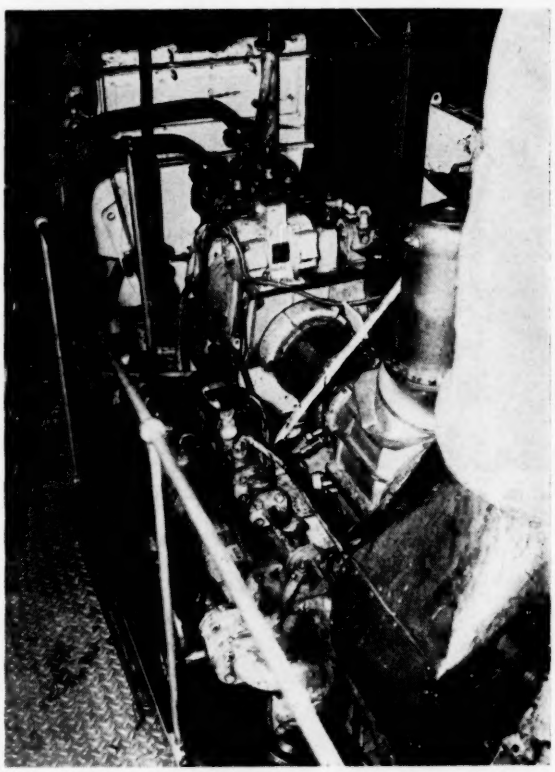


Fig. 21 500-kw generator set driven by Allen gas turbine Type C, installed in *HMS Llandaff* (Courtesy British Admiralty)

Table 11 Gas turbines in naval auxiliary applications

Country	Manufacturer of engine	Model	Power/ plant	Plants/ ship	Ship	No. of ships	Status	Service
England	W. H. Allen Sons	Type C	500 kw	1	*HMS Llandaff*	5	Building	Generator
England	W. H. Allen Sons	Type C	500 kw	2	...	2	Building	Generator
England	Rover Gas Turbines, Ltd.	1S/60	40 kw	2	*HMS Brave* class	2	1 built	Generator
England	Ruston & Hornsby	TA	750 kw	1	*HMS Cumberland*	1	Installed	Generator
U. S.	Boeing Airplane Co.	502-10C	220 hp	1	MSL	28	Installed	Generator
U. S.	Solar Aircraft Co.	T-520	500 hp	2	MSB-28	2	Installed	Generator
U. S.	Solar Aircraft Co.	T-520	300 kw	1	*USS Gyatt*	1	Installed	Generator
U. S.	Solar Aircraft Co.	T-45	46 hp	1	Beach lighter	1	Installed	Fire pump
U. S.	Solar Aircraft Co.	T-45	46 hp	1	PT	1	Installed	Fire pump
U. S.	Solar Aircraft Co.	T-45	46 hp	1	Various	36	Installed	Fire pump
U. S.	Solar Aircraft Co.	T-45	46 hp	1	...	62	Installed	Auxiliary
U. S.	Solar Aircraft Co.	T-45	46 hp	1	LCU	13	Ordered	Fire pump, fog
U. S.	Boeing Airplane Co.	502-6	160 hp	4	MSB-5	46	Installed	Generator
U. S.	Boeing Airplane Co.	502-8C	160 hp	1	MSL	2	Installed	Generator
U. S.	Boeing Airplane Co.	500-10C	...	1	LST	7	Installed	Fog

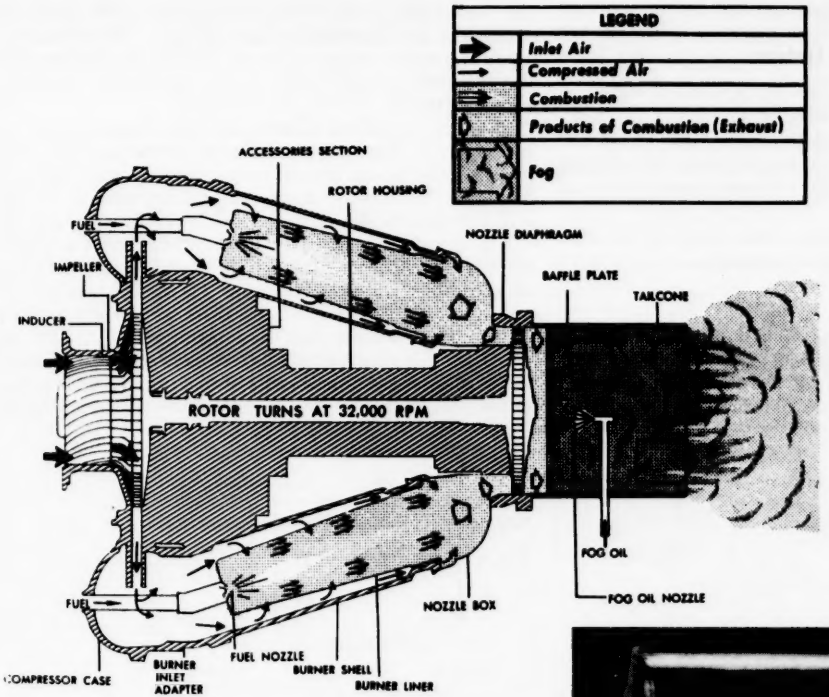


Fig. 22 Fog generator, using gas turbine, installed on seven LST's

Rotor speeds are 19,450 and 26,550 rpm for the l-p and h-p rotors, respectively. Design turbine-inlet temperature was 1292 F with a specific fuel consumption of 0.952 lb per kwhr. The design and development information relating to this engine was presented to The Institution of Mechanical Engineers in London, January 10, 1958 [38].

2 *300-kw Generator.* The U. S. Navy installed a 300-kw emergency and stand-by generator set, Fig. 4, Table 11, on the *USS Gyatt* in November, 1956. The generator characteristics are: 300 kw, 450 volts, 3 phase, 60 cycle, 0.8 power factor. The gas turbine is a Solar T-520J, single-shaft machine, 10-stage axial-flow compressor, pressure ratio 4.6:1, 3-stage axial-flow turbine. Specific fuel consumption is 0.96 lb per hp-hr at full load and 1.35 lb per hp-hr at 50 per cent load. Starting is accomplished by a combustion-air starter. It is required to start from a cold condition and carry full load within a period of 10 sec. The installed weight of the turbine and generator is 8000 lb. Inlet and exhaust ducting for the entire installation weighs only about 1200 lb. This is about 9500 lb lighter than a diesel-driven generator set of the same capacity [8].

3 *Fog Generators.* The U. S. Navy has recently installed Boeing gas-turbine fog generators on seven new LST's. The gas turbine is well suited for this purpose. Fog is generated by spraying fog oil into the turbine exhaust of the gas-generator portion of a gas turbine as shown in Fig. 22. This unit is remote controlled, operated by a single push button, weighs 650 lb, occupies only 12 sq ft of deck space, and can produce fog within 30 sec without prior warm-up.

4 *Fire Pump.* The gas-turbine-driven fire pump is a very compact, reliable, and easily moved unit. The U. S. Navy has procured a number of Solar T-45 fire pumps, Fig. 15, for use in ships and small craft. The pump and its driving turbine are mounted in a 24 × 26-in. tubular frame. The package weight is about 180 lb. The pump delivers 500 gpm at a pressure of 100 psi. The turbine is rated at 46 hp. The Rover 1S/60 (Curtiss-Wright 300 ST) gas-turbine-driven fire pump, Fig. 23, is rated at

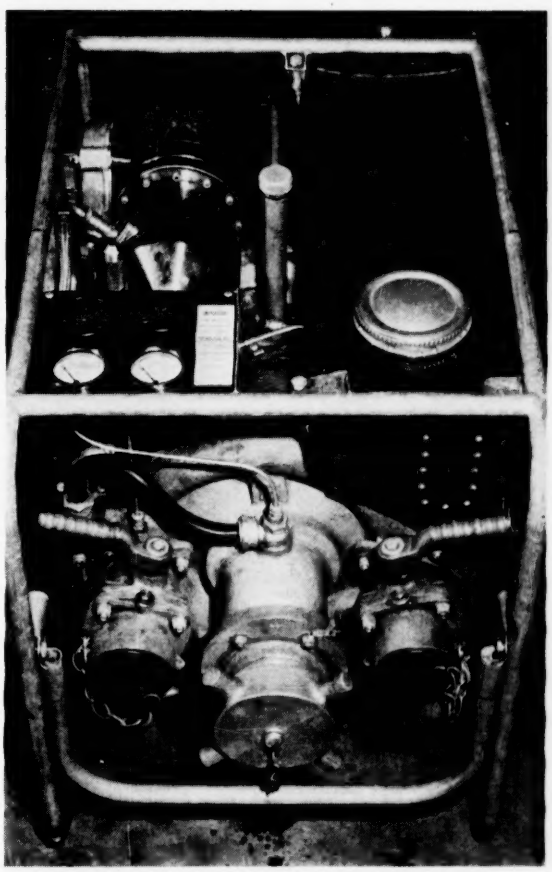


Fig. 23 Gas-turbine-driven fire pump, 500-gpm capacity at 100 psi, Rover 1S/60 (Curtiss-Wright 300 ST) (Courtesy Curtiss-Wright Corporation)

500 gallons per minute at a discharge pressure of 100 psi.

Requirements for Marine Gas Turbines

The ultimate requirement of a prime mover is that it perform a given task at a minimum cost, whether in naval or merchant service. This applies equally well to auxiliary and propulsion applications. A generalization of requirements for the gas turbine in marine use suggests the following points for consideration:

1 Reliability
2 Cost (initial, maintenance, operation, overhaul, fuel)
3 Ability to meet all operating requirements
4 Weight
5 Space

There are two general types of gas turbines in marine use. One is of the long-life design with good over-all thermal efficiency where reduction of weight and space are not of paramount importance. Duct velocities are kept low in order to reduce losses. A second type incorporates more of the aircraft concept, with particular emphasis on the reduction of weight and space. Cycle temperatures are generally higher than in those of the long-life type and duct velocities are usually high in order to keep duct sizes to a minimum [16].

Merchant applications to date have been limited almost exclusively to the heavier-type machines. In the propulsion field both the initial plant of 1200 hp and the new 5500 hp *Auris* plants, and the 6000-hp *John Sergeant*, are well known. Specific weight of the *John Sergeant* is about 45 lb per shp, including turbine, gear, and regenerator [16]. The 1200-hp *Auris* plant had a specific weight of about 102 lb per shp, including total weight of the gas turbine and alternator [39]. The specific weights on a lb-per-bhp basis are shown in Table 12 for the new and old *Auris* plants. The new *Auris* plant has a specific weight of 68.3 lb per bhp [54]. Generator sets installed on merchant ships have specific weights for gas turbine and gear similar to the 8.1 lb per shp for the Allen 500-kw set [38]. Small auxiliary engines such as the

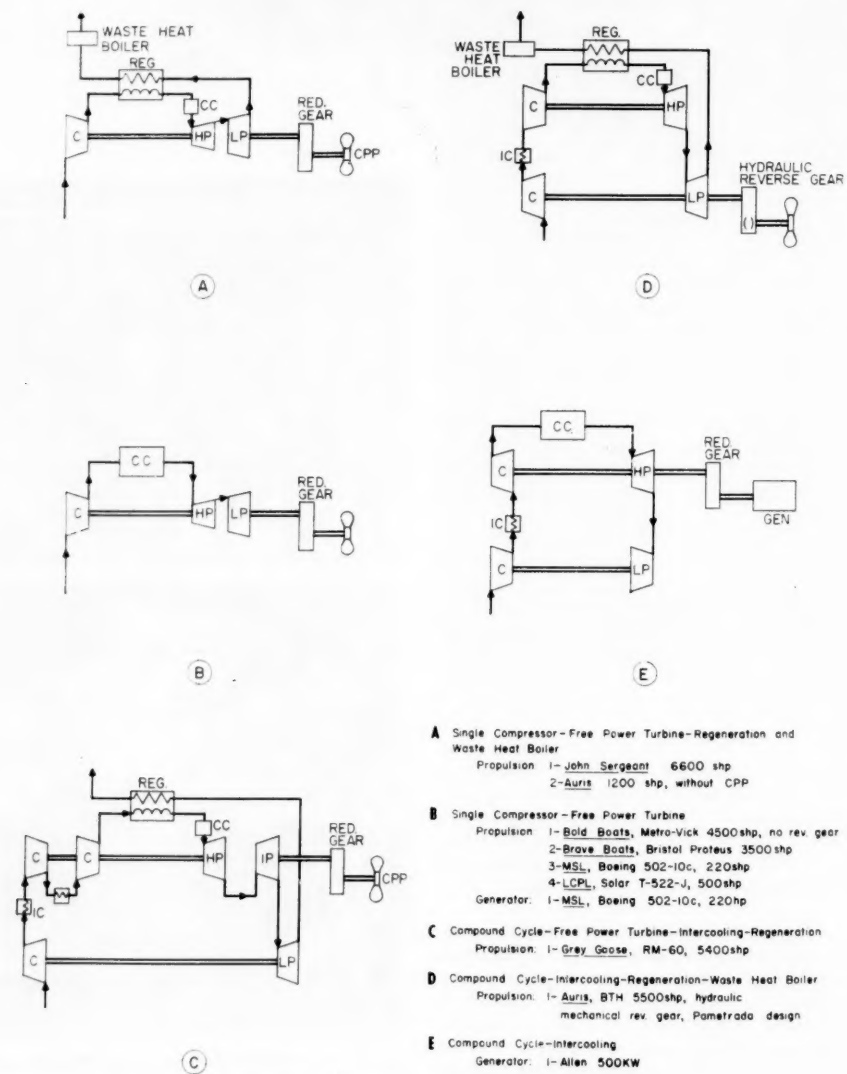


Fig. 24 Basic cycles for marine gas turbines

Solar T-45 fire pump installed on the U. S. Maritime Ship *Thomas Nelson* have specific weights of about 4.0, including gears, pump, controls, and mounts.

Naval uses to date have been concerned with the lightweight compact engine with corresponding shorter overhaul periods. The Proteus marine engine, Fig. 5, installed in *HMS Brave Borderer*, Fig. 19, has a specific weight of 1.6 lb per shp [34]. This engine has a 1500-hr overhaul interval [9]. The 300-kw generator set on the *USS Gyatt* has a total installed weight of 8000 lb. The specific weight of the gas turbine and reduction gear on this set is about 2.1 lb per hp [8]. As for any other prime mover the ultimate selection of a gas turbine will be the result of logical compromises in order to get the optimum plant for the particular application.

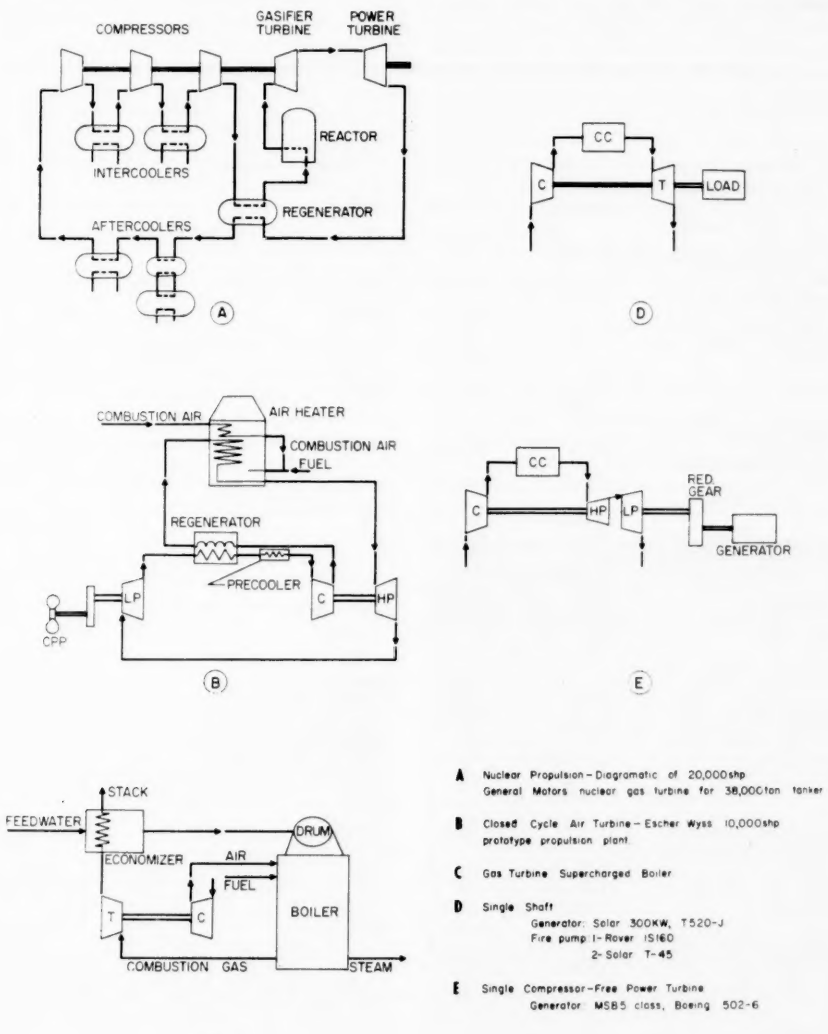
Cycles for Marine Gas-Turbine Plants

Marine gas turbines installed to date have been open cycle. All installations in the United States are open-cycle without regeneration except the *John Sergeant*. British installations have also followed the open cycle. However, some incorporate compound engines with intercooling and regeneration. The basic cycles now in use at sea are presented in Figs. 24 and 25. The simplicity of the open-cycle plant accounts for its extensive use up to the present.

A prototype closed-cycle plant is now under construction by Escher-Wyss [36], and the closed-cycle gas turbine is considered essential for the success of the nuclear gas-turbine-propulsion system. This has been the subject of many investigations during the past few years, including the plant which the U. S. Maritime Administration has under development by The General Dynamics Corporation.

Developments

The relatively large increase in marine-gas-turbine horsepower, 2500 in 1947 and approximately 313,000 in 1957, has resulted from new developments and their acceptance in the marine-engineering



A Nuclear Propulsion – Diagramatic of 20,000shp General Motors nuclear gas turbine for 38,000ton tanker

B Closed Cycle Air Turbine – Escher Wyss 10,000shp prototype propulsion plant

C Gas Turbine Supercharged Boiler

D Single Shaft
Generator: Solar 300KW, T520-J
Fire pump: 1- Rover 1S160
2- Solar T-45

E Single Compressor–Free Power Turbine
Generator: MS85 class, Boeing 502-6

Fig. 25 Basic cycles for marine gas turbines

field. Gas turbines and free-piston gas turbines in marine service are shown in Tables 4 and 13. Gains to be realized are perhaps not as great as in the aviation industry which was forced to the gas turbine for military use, and is rapidly changing over to the turboprop and turbojet for commercial use [16]. The increase, while somewhat impressive, is not as large as might be expected. The problem of getting complete acceptance of the gas turbine is still of major proportions. The impressive low-operating-cost record such as that obtained by Shell in the gas-turbine electric-generating plant at La Concepción, Venezuela, is most promising. There, the operating and maintenance cost per kwhr of the gas turbine was only half that of the diesel [13], [40]. Experience such as this, and that of railways and aircraft, indicates lower operating costs for the marine industry. Improvements have been made in mechanical and aerodynamic design, materials, manufacturing techniques, and operating methods based on aviation, marine, and industrial experience. With the proper selection and combination of desirable features from many fields, gas turbines for marine use will be further improved in performance and reliability.

A Marine Gas Turbines Available. A number of new marine engines are available or under development, a large proportion of which are described in "Marine Gas Turbine Engine Data Book," prepared by the Bureau of Ships, Department of the Navy, Wash-

Table 12 Comparison of *Auris* gas-turbine propulsion plants [54]

Data	First plant	New plant
Net output, bhp	1300	5500
Ambient air, deg F	68	75
Pressure ratio, total	4.0	6.1
Intercooler thermal ratio, per cent	none	84
Turbine inlet, deg F	1108	1200
High-pressure turbine, rpm	6000	5800
Low-pressure turbine, rpm	2870	3840
Specific fuel consumption, design point, lb/bhp-hr	0.64	0.505
Over-all thermal efficiency, per cent	22.1	28.0
Specific weight (includes turbine, frames, ducts, excludes gears), lb per bhp	86.2	68.3
Starting means	electric motor	steam

Table 13 Characteristics of marine gas turbines installed or building, 1953 through 1957[d]

Country	Manufacturer	Model	Continuous power	Description[a]	Number of engines	Weight including gear, lb	Turbine-inlet temperature, deg F	Pressure ratio	Specific fuel consumption, lb per shp-hr
England	W. H. Allen Sons & Co., Ltd.	Type C	500 kw	SPT	9	11200	1380	10.0:1	1.1[b]
England	W. H. Allen Sons & Co., Ltd.	Type B-1	125 kw	SS	1	750	1300	2.5:1	2.3[b]
England	W. H. Allen Sons & Co., Ltd.	Type D	350 kw	SS	6	3000	1380	2.7:1	1.93[b]
England	British Thomson-Houston	...	1300 hp	SPTR	1	112000	1108	4.0:1	0.64[c]
England	British Thomson-Houston	...	5500 hp	PTL-R-I-W	1	376000	1200	6.0:1	0.505[c]
England	Blackburn	Turmo 600	350 hp	SPT	2	310	1500	4.0:1	1.15
England	Metro-Vickers Electric Co., Ltd.	G2/1	4500 hp	SPT	4	9700	1472	4.0:1	0.80
England	Metro-Vickers Electric Co., Ltd.	G2/2	4000 hp	SPT	2	9700	1436	4.0:1	0.85
England	Rover Gas Turbines, Ltd.	1S-60	40 kw	SS	..	...	1454	2.9:1	...
England	Ruston Hornsby, Ltd.	T-A	1300 hp	SPT	1	12600	1340	4.0:1	0.91
England	Ruston Hornsby, Ltd.	T-E	300 kw	SS	2	4400	1470	3.5:1	0.98
England	Rolls-Royce Ltd.	RM-60	5400 hp	SPT R-I	2	26000	1510	18.0:1	0.66
France	Rateau	ACB	3300 hp	I-R	1	...	1290	9.0:1	0.49
Japan	Mitsubishi	...	2500 hp	SPT-IRH	..	77000	1290	10.0:1	0.50
Japan	Mitsubishi	EWJN10	10000 hp	Closed-cycle, IR	2	325000	1340	3.8:1	0.50
Japan	Mitsubishi	...	200 hp	SPT	..	2100	1472	3.7:1	1.20
U. S.	Boeing Airplane Co.	502-2	160 hp	SPT	1	236	1550	3.0:1	1.35
U. S.	Boeing Airplane Co.	502-6	160 hp	SPT	184	315	1550	3.0:1	1.40
U. S.	Boeing Airplane Co.	502-8C	160 hp	SPT	6	319	1580	3.0:1	1.37
U. S.	Boeing Airplane Co.	502-10C	220 hp	SPT	58	325	1535	4.0:1	1.06
U. S.	Boeing Airplane Co.	Fog generator 500-10C	...	SS	7	650	1500	4.0 1	...
U. S.	General Electric Co.	...	6600 hp	SPT R-W	1	...	1450	4.9:1	0.51
U. S.	Pratt & Whitney Aircraft	T-34	5000 hp	SS	4	2590	...	6.7:1	0.74
U. S.	Solar Aircraft Co.	T-520	500 hp	SS	5	850	1450	4.6:1	1.10
U. S.	Solar Aircraft Co.	T-522	500 hp	SPT	4	1000	1450	4.6:1	1.10
U. S.	Solar Aircraft Co.	T-45	46 hp	SS	95	65	1300	2.4:1	2.0
U. S.	Lycoming Div. of AVCO	T-53	685 hp	SPT	1	480[e]	1500	6.0:1	0.75

[a] PTL = power turbine driving low-pressure compressor, R = regenerator, I = intercooled, H = reheat, W = waste-heat boiler, SS = single-shaft, SPT = separate power turbine.
[b] Lb of fuel per kwhr.
[c] Lb per bhp-hr.
[d] Not including spare engines.
[e] Less reduction gear.

ington, D. C. The 1956 issue [41] listed 13 manufacturers and 42 engines. The 1958 edition [42] lists 16 manufacturers and over 60 engines. The manufacturers and engines, with some data, are shown in Table 8. The increase in the number of engines now available is a further indication of progress being made in marine applications of gas turbines.

B Engines Under Development. Engines that are in various stages of development include propulsion, base-load, boost, and auxiliary applications. Brief descriptions of some of these engines follow:

1000-hp. This is a simple-cycle engine with a free-power turbine. It is being developed by Solar Aircraft primarily for high-speed boat propulsion. However, there may be other uses for this extremely compact, lightweight, and low-specific-fuel-consumption engine. The engine weight is about 725 lb. Dimensions are approximately 62 × 42 × 42 in. (length, width, height). Design pressure ratio is 6.5:1 with an air mass flow of 12.5 lb per sec. Turbine-inlet temperature is 1475 F with a specific fuel consumption at full load of 0.65 lb per shp-hr. The overhaul interval is 2000 hr. The gas-producer portion of this engine first went on test in December, 1957. From these early results the design goal of 0.65 specific fuel consumption is expected to be realized with little difficulty.

7500-hp. This is an extremely lightweight engine, Fig. 26, developed by Westinghouse for the Bureau of Ships for use as a boost unit in conjunction with a steam-turbine base-load plant. Continuous rating is 7500 hp. The engine dry weight is 8075 lb. It has an axial-flow compressor with a pressure ratio of 4.45:1 and an air mass flow of 130 lb per sec. Specific fuel consumption is 0.86 at full power. Turbine inlet temperature is 1350 F. Engine size is 18 × 7.5 × 6.5 ft. Factor test of 500 hr at full power was completed satisfactorily on one engine. Three additional engines have passed acceptance tests. Four engines have now been delivered. Three are now on test at the U. S. Naval Boiler and Turbine Laboratory in combination with a steam plant, Fig. 27.

3500-hp. A lightweight engine is currently under development by Curtiss-Wright, Utica Division, for the Bureau of Ships for use as a propulsion or boost-power unit. It would also have potential as an auxiliary drive for an air compressor. This is a simple open-cycle machine with a free-power turbine. The design calls for a 7.25:1 pressure ratio, air mass flow 43 lb per sec, specific fuel consumption of 0.65 lb per hp-hr, turbine-inlet 1540 F, with a weight of 6800 lb.

7-kw Generator. Three portable generator sets for field use are being procured from Solar Aircraft by the Bureau of Ships.

600-kw Generator. The U. S. Maritime Administration has solicited bids on a 600-kw gas-turbine-generator set for a ship installation.

300-kw Generator. An exceptionally compact package generator set, Fig. 36, with the turbine placed over the generator, has been designed and built for the U. S. Navy and will be tested in the near future.

Astern-Power Methods. The question of supplying astern power in gas-turbine propulsion plants was a serious one a few years ago, particularly, above several thousand horsepower per shaft. A number of methods have been proved at sea and others are under development. These include: (*a*) Reverse reduction gear, mechanical; (*b*) hydraulic reversing couplings; (*c*) controllable propeller; (*d*) electrical transmission; (*e*) astern stages in the main-turbine casing; (*f*) radial-inflow turbine with reversing nozzles.

Reverse Reduction Gear, Mechanical. This method of obtaining astern power has been widely used on numerous gasoline and diesel-powered small craft by both merchant and naval services for many years, and reverse gears up to 3500 hp are in use on gas-turbine boats. The British Navy recently launched the *Brave Borderer* which is equipped with a reverse reduction gear, manufactured by W. H. Allen Sons and Company, Ltd., for the 3500-shp Proteus marine engine [34]. Numerous small craft such as the 200-shp MSL, 500-shp LCPL, and 200-shp LCVP have used commercial-type reverse gears. Experience with a commercial reverse gear in a U. S. Navy 40-ft Personnel Boat showed that the boat could be brought to a dead stop in 9 sec from about 25 knots full speed ahead. The ahead reach was 110 ft. The same boat was brought to a stop in one half its length from an ahead speed of 10 knots [35].

Hydraulic Reversing Couplings. This system, developed by Pametrada and installed in the new 5500-shp propulsion plant for the *Auris*, Fig. 28, consists of ahead and astern hydraulic couplings constructed in the main reduction-gear housing. The turbine-output coupling is connected to a flexible coupling which in turn is attached to a quill shaft. The driving wheels of the ahead hydraulic coupling and the astern torque converter are keyed to the quill shaft. The ahead coupling is at the forward end of the quill shaft, and the astern torque converter at the after end. The driven wheels of the ahead coupling and the astern converter are fixed to the high-speed pinion. The astern converter, Figs. 38 and 39, has a fixed-blade cascade between the driving and driven

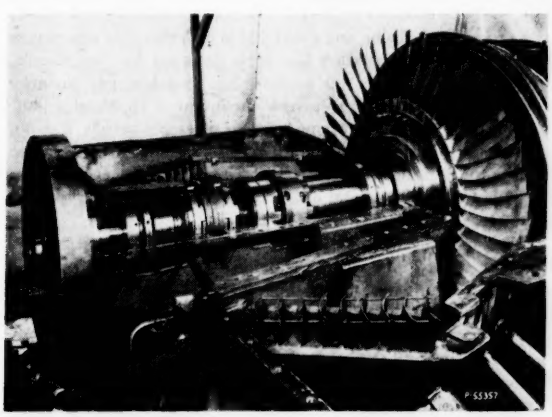


Fig. 26 Power turbine of a 7500-hp boost gas turbine for marine propulsion (Courtesy U. S. Navy)

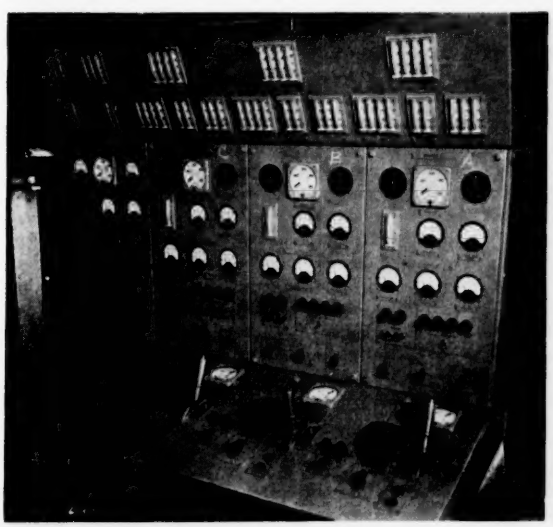


Fig. 27 Control console for Cosag gas turbines on test at U. S. Naval Boiler and Turbine Laboratory (Courtesy U. S. Navy)

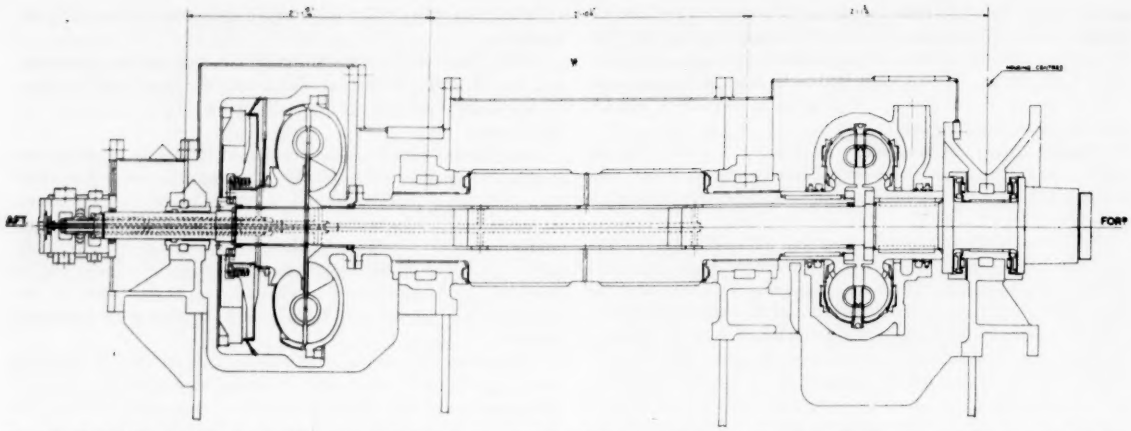


Fig. 28 Hydraulic-mechanical reversing gear installed in the 5500-hp *Auris* propulsion plant (Courtesy the Parsons and Marine Engineering Turbine Research and Development Association)

wheels to give astern rotation. Ahead or astern operation is accomplished by filling the proper coupling with oil. The oil supply is controlled by a single piston valve which is interlocked with the fuel control to assure proper operation. The efficiency of the ahead coupling is about 98 per cent. Under full-power-ahead operation, a direct-drive hydrocentrifugal friction clutch will be used to eliminate the 2 per cent slip in the hydraulic coupling. Astern converter efficiency is about 65 per cent. Studies have been made to determine the suitability of this maneuvering system to a 12,500-shp ship with a propeller speed of 112 rpm. The hydraulic system was compared with the astern stages in a steam turbine. Time to stop the propeller for the astern turbine and the hydraulic system was 11.5 and 5.0 sec, respectively. The ship-stopping time for the astern turbine and the hydraulic system was 5.5 and 5.0 min, respectively [16, 43, 44, and 45].

Controllable Propellers. Considerable experience has been gained with the controllable propeller in marine-propulsion plants [46]. Gas-turbine installations with controllable propellers vary in power from 180 to 6600 shp. Designs are in progress for a 10,000-shp controllable propeller [36] and the U. S. Navy has a development project for a 20,000-shp unit [47]. Some recent gas turbine installations with controllable propellers are: (*a*) *Grey Goose*, 5400 shp (Rotol) [10]; (*b*) MSL, 180 shp [33]; (*c*) *John Sergeant*, 6600 shp (S. Morgan Smith), [25]; (*d*) PT-812, 600 shp.

Electrical Transmission. The 1200-hp gas turbine driving an alternator in the *Auris* is an example of the electrical system. The results of this operation have been satisfactory.

Astern Stages in Main-Turbine Casing. This method is similar to the steam-turbine practice of incorporating astern stages in the ahead casing. The windage losses and overheating during ahead and astern operation are not prohibitive. In the gas turbine overheating and casing distortion have been a problem where astern stages are used. Limited astern power can be developed by this means. The *William Patterson* has two astern rows in each of two 3000-shp gas turbines. Each turbine develops about 1100 shp astern. Ships with free-piston gas-turbine plants with astern stages include: 21 French minesweeps; *William Patterson, Ormara;* two French coastal ships; three fishing trawlers; and an ore carrier.

Radial-Inflow Turbine With Reversing Nozzles. Development of reversing nozzles, Fig. 29, for use with a radial-inflow turbine is being conducted by the AiResearch Corporation for the Bureau of Ships. Nozzles have been built and tests are now being made to determine performance of a turbine under ahead and astern operation. This method of reversing when fully developed may find many applications in merchant and naval service. It offers some attractive features for a closed-cycle plant where high power can be developed by a relatively small radial-inflow turbine.

Liquid Cooling. Work on liquid cooling of turbine blades is continuing. The closed system, developed by Solar Aircraft Company for the Bureau of Ships, in which water is circulated through blade passages and then out of the engine to a heat exchanger, has been operated at 1700 F for more than 100 hr [48]. The Pametrada liquid-metal-cooled turbine, Fig. 30, where NaK is sealed in each blade, has been operated at 2100 F. Significant improvements can be realized by cooling, which, in turn, permits use of higher cycle temperatures. Thermal efficiencies greater than 40 per cent with cycle temperatures of 2200 F are indicated [16].

C Combined Plants. Gas turbines have been combined with both steam turbines and diesels in marine propulsion plants. The existing installations are naval. The gas turbine combined with steam turbine is referred to as Cosag, Fig. 31. When gas turbine and diesel are used for propulsion the plant is called Codag [6]. Codag plants include such well-known vessels as *Auris, Bold Pathfinder,* and *Bold Pioneer.* The *Auris* combined plant provided early sea experience on a gas turbine while having 75 per cent of the propulsion plant as diesel. The British Navy Bold-class boats utilized diesels for cruising and maneuvering while the gas turbines were designed to be used for high boat speeds [10]. The U. S. Navy's PT-812 has gas turbines for high speed, with diesels for cruising and controllable propellers for maneuvering. A marine installation has been designed for two diesels, providing cruise power up to 4000 hp, and two gas turbines coupled with the diesels to give a maximum of 16,000 shp [49]. The German Navy is now constructing six escort vessels, Fig. 40, with combined gas-turbine-and-diesel propulsion. Diesels are to be used for cruising with gas turbines supplying the peak power. Each ship will have a total of 38,000 hp. The first ship is to be completed in 1959 [67].

Cosag propulsion, like Codag, offers the greatest attraction in naval applications where demands for extremely high speeds are required during relatively short periods. The U. S. Navy has developed a 7500-hp gas-turbine engine, Fig. 26, for use with a steam plant. This engine is now being tested along with a steam turbine at the U. S. Naval Boiler and Turbine Laboratory. The British Navy has selected the Cosag-type plant for a guided-weapon destroyer and a small frigate [50]. Here the gas turbine will supply the peak power with the steam turbine serving as the base-load plant.

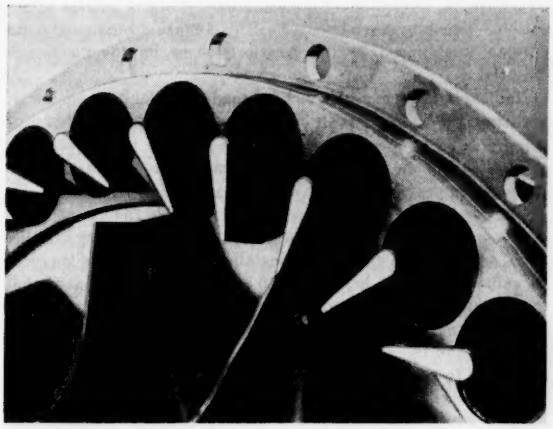


Fig. 29 Reversing nozzles in a radial-inflow gas turbine (Courtesy AiResearch Manufacturing Company)

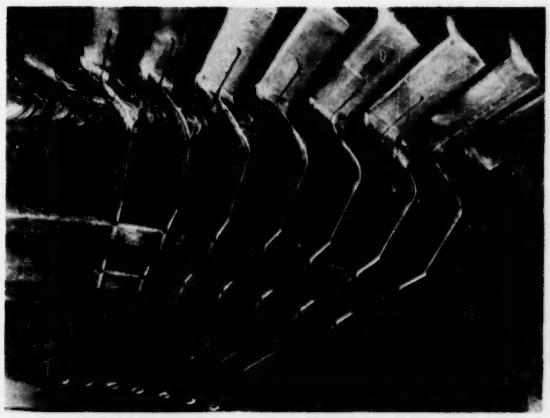


Fig. 30 Pametrada liquid-cooled gas-turbine rotor grooved for thermocouples (Courtesy The Parsons and Marine Engineering Turbine Research and Development Association)

D Supercharger for Pressure-Fired Boiler. Attractive savings in weight and space can be realized by use of the supercharged boiler in the marine propulsion plant. A supercharged-boiler cycle is shown in Fig. 25. Studies conducted on a 6000-shp supercharged boiler for a merchant ship show a weight reduction of 88 per cent and a volume reduction of 80 per cent from the boilers used with a 2500-shp reciprocating steam engine [51]. An axial-flow supercharger has been manufactured for the U. S. Navy by the Elliott Company for a pressure-fired boiler. This supercharger has the following design characteristics: 11-stage axial-flow compressor; capacity 40,000 cfm; pressure ratio 4.5; inlet conditions 14.7 psia at 100 F; power required 6100 hp. The gas turbine is a two-stage axial-flow unit with inlet gases at 61.2 psia and 815 F [52]. This supercharger is now being tested with a boiler at the U. S. Naval Boiler and Turbine Laboratory.

E Free-Piston Plants. Considerable attention is being focused on the free-piston turbine for propulsion in England and on the continent. This prime mover is also being developed for electric-generator drive. Marine installations of this prime mover, which were mentioned earlier, involve some 107,000 hp and 38 vessels, Table 1. The construction of two new 16,000-shp merchant ships for La Marine Nationale Française indicated that the free-piston gasifiers are of major interest to the European ship operators [30]. The problem of astern power is under study by many groups. Methods in use include: Astern stages in the main-turbine casing; controllable propellers; mechanical reversing gear; hydraulic reversing gear; separate ahead and astern turbines; and electrical transmission systems. Free-piston-gasifier propulsion plants are installed or building for some 30 ships totaling about 70,000 shp, with astern stages in the propulsion turbine. Two ships are building with controllable propellers, totaling about 32,000 shp.

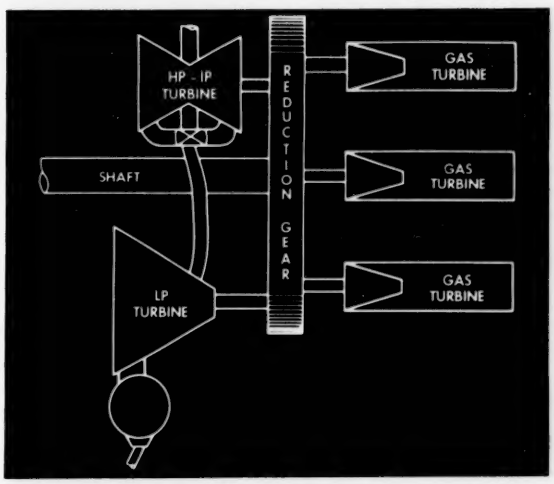


Fig. 31 Gas and steam turbines combined to form a propulsion plant (Courtesy U. S. Navy)

A 200-kw generator set, driven by a CS-75 gasifier and a Budworth gas turbine, has been completed. This is a prototype of sets being manufactured for the New Zealand Shipping Company, Ltd., and the Port Line, Ltd. [31]. The total weight of the set (dry) is about 14,000 lb.

Methods for increasing power output of gasifiers were discussed in a recent paper [30]. These include: (*a*) Supercharging the gasifiers by precompressing the air; (*b*) after-burning between gasifier and turbine; (*c*) twinning (Siamese) of gasifiers,

Table 14 Relative weights of a 20,000-shp propulsion plant using various nuclear systems [53]

Cycle	CO_2	Helium	CO_2-Air	CO_2-CO_2	Pressurized water
Propulsion machinery	300,250	305,880	304,600	302,300	512,000
Reactor plant	914,070	907,050	1,331,490	1,187,350	609,000
Shielding[a]	1,133,500	974,500	945,500	945,500	1,691,000
Electric plant	176,500	176,500	310,000	310,000	296,000
Independent system machinery	85,900	85,900	85,900	85,900	186,000
Total weight, lb	2,610,220	2,449,830	2,977,490	2,831,050	3,294,000
Weight-power ratio, lb/shp	131	122	149	141	165

[a] These shielding weights are based on the use of liquids for secondary shielding; for each of the cycles listed a weight credit of 520,000 lb has been taken for diesel fuel oil used also as shielding.

common engine case for scavenging air; (*d*) making gasifiers with larger bore and stroke.

The advantages and disadvantages of these methods were explored in detail. It appears that these new developments in marine applications as well as improvements in gasifiers are combining to further increase the use of the free-piston turbine afloat.

F Nuclear Gas Turbines. One of the most controversial subjects in the marine field has been the nuclear-gas-turbine propulsion plant. The most significant step taken to date on this subject was the decision in the United States of the Atomic Energy Commission, and the Maritime Administration, United States Department of Commerce, to develop a prototype gas-cooled reactor, closed-cycle gas-turbine plant for merchant-ship propulsion, to develop 20,000 hp, announced in a January 23, 1958, press release. Relative weights of the propulsion plant for various cycles are shown in Table 14. This tabulation shows all gas-turbine cycles have a marked advantage over the pressurized water in weight [53].

G An Experimental Propulsion Plant. Four Pratt & Whitney 5000-hp T-34 gas turbines driving air propellers were installed on a Liberty hull, Fig. 37. Maneuvering was done by training the engines as required.

Operating Experience at Sea

The normal sequence in development and ultimate widespread use of the gas turbine at sea has been no different from any other "new" concept. The marine gas turbine has had many ardent supporters. It has also had many opponents who have thrown out some serious challenges. In the very early stages of development and sea operation there were instances of "overselling." These, coupled with mechanical problems and relatively high fuel consumption, naturally resulted in a slow-down of marine application. This may have been a healthy condition because the gas turbine was placed in the position where improvement was essential.

Radical improvements have been made in reliability, economy, and simplicity. Leaders in the marine field recognize that this prime mover, in many applications, is superior to reciprocating engines and steam turbines. The increase of about 310,000 hp in the past ten years, Table 1, is a real measure of progress.

The gas-turbine installations covered in Table 15 have accumulated about 103,000 hr at sea on propulsion and auxiliary installations. Free-piston gas-turbine installations at sea have operated an estimated 72,820 hr, Tables 16 and 22. It is estimated that ships and boats have been driven over 314,000 miles by gas turbines and more than 500,000 miles by free-piston gas turbines, Tables 15 and 16.

A Gas Turbines. 1 Propulsion Gas Turbines. Propulsion gas turbines represent some 255,000 hp installed in 106 ships and boats, Table 1, including both merchant and naval usage. Some general discussions of the operating experiences on representative vessels follow:

Auris [*54*]. The 1200-hp gas turbine, installed in this ship in October, 1951, was in service for 5 yr. The total running time was 20,510 hr which includes 679 hr on the manufacturer's test stand. Marine diesel or gas oil was burned for 13,861 hr. During the remaining 6649 hr 1500-sec Redwood-I-viscosity 100-F residual fuel was used. The 1200-hp turbine, along with three 1100-hp Sulzer diesels have been removed from the ship and replaced with a 5500-hp gas turbine. A comparison of the two plants is shown in Table 12. During the 5-yr operation ambient temperatures varied from a high of 103 F to a low of 21 F. The gas turbine was running during 179,775 nautical miles traveled.

(*a*) *Down Time.* During the *Auris* operation there were 15 in-

Table 15 Operating hours at sea on gas turbines[a]

Vessel	Country	Engine manufacturer	Rating[c]	Application[b]	Hours operation[d]	Operation up to	Vessel miles[e]
Auris	England	British Thomson Houston	1200	MP	19831	12/55	179775
John Sergeant	U. S.	General Electric Co.	6600	MP	4723	4/58	62000
HMS Grey Goose	England	Rolls-Royce	5400 × 2	NP	1344 1279	1/57	
HMS Bold Pathfinder	England	Metropolitan Vickers	4500 × 2	NP	491 471	12/57	
HMS Bold Pioneer	England	Metropolitan Vickers	4500 × 2	NP	355 347	12/57	
HMS MGB 2009	England	Metropolitan Vickers	2500 × 2	NP	550	12/57	
HMS Llandaff	England	W. H. Allen	500 kw	NG	30	12/57	
Dolius (5 ships in class)	England	W. H. Allen	350 kw	MG	600	4/58	
HMS Cumberland	England	Ruston & Hornsby	750 kw	NG	1400	12/57	
USS Gyatt	U. S.	Solar Aircraft	300 kw	NG	80	4/58	
Minesweep MSB (2 boats)	U. S.	Solar Aircraft	200 kw	NG	3521	4/58	
Landing craft (1 boat)	U. S.	Solar Aircraft	500	NP	275	4/58	
Personnel boat—40 ft	U. S.	Solar Aircraft	500	NP	800	4/58	16000
PT-812	U. S.	Metropolitan Vickers	4000 × 2	NP	62	12/57	
Minesweep MSB (46 boats)	U. S.	Boeing Airplane	160 × 4	NG	65083	4/58	
Minesweep MSL (2 boats)	U. S.	Boeing Airplane	160	NG	180	12/57	
Minesweep MSL (4 boats)	U. S.	Boeing Airplane	160	NP	1525	4/58	
LCVP (3 boats)	U. S.	Boeing Airplane	220	NP	120	12/57	
Minesweep (62 boats)	U. S.	Solar Aircraft	46	NA	906	4/58	
LCVP (1 boat)	U. S.	Boeing Airplane	160	NP	205	12/57	
Personnel boat (Creole)	U. S.	Boeing Airplane	220 × 2	MP	204 204	4/58	
				Total	103000 (approximately)		314000 (approximately)

[a] Some recent installations not in this summary.
[b] P = propulsion, G = generator, A = air supply, U = pump, M = merchant, N = naval.
[c] Distance vessel traveled with propulsion gas turbines in operation.
[d] Where two groups of hours appear on one vessel, first is port and second is starboard-engine propulsion.
[e] Rating shown in horsepower unless otherwise specified.

stances when the turbine was not available for use at sea, ranging from 2 hr to 73 days. The causes of engine shutdown and remedial action taken are summarized in Table 17. It is important to appreciate that this was an *experimental* installation, the *first* commercial plant of any size, and many new features were in process of development.

(*b*) *Compressor.* This component gave little trouble. Lubricating-oil vapors were drawn into the compressor from the pedestal-bearing labyrinth seal. A new type seal will be on the new turbine. Removal of salt deposits from compressor blades was accomplished by dumping 10 lb of flour-mill by-products, "ground screenings," into the compressor inlet at full power. This restored compressor performance in about one minute. Injection of fresh water and a detergent was tried but this could only clean the unit when operating at 400 rpm, which required a shutdown.

(*c*) *Heat Exchanger.* Water-washing to remove soot was first used. Soot blowers were installed and they were operated once every 24 hr. This eliminated the need for any further water-washing.

(*d*) *Combustion Chambers.* Life of these components was very good. Major changes included the replacement of mixing-zone shells with Nimonic-75 and fire bricks of 80 per cent alumina were installed in place of the 70 per cent alumina. Combustion with all fuels was excellent.

(*e*) *High-Pressure Turbine.* During 20,000-hr operation this turbine caused three shutdowns for a total of 160 hr. Contaminated fuel caused one shutdown and minor mechanical defects contributed two shutdowns. Thermal stresses caused cracks to develop in the top half of the casing. Casing distortion was also evident when the turbine was opened after 17,510 hr. Blade-tip rubs had occurred on rows 1 and 5. Clearances between high-pressure flanges, upper and lower, unbolted, varied from 0.005 to 0.112 in. However, there was no gas leakage when bolted down. Dimensional checks on the rotor after 17,000 hr showed it was running true to within 0.0035 in. The deposition and corrosion of blades, when burning high-viscosity fuels, were the greatest problems encountered with the h-p turbine. Mechanical damage was evident on the stator and rotating blades. Some of this was due to carry-over of firebrick retaining lugs. Blade-tip rubs also accounted for some damage. Based on the facts determined after inspections, the turbine appeared to have a useful life of at least 20,000 hr.

(*f*) *Fuel.* Specification for the *Auris* only required a viscosity not to exceed 1500 sec Redwood I at 100 F. A fuel used during one series of tests is described in Table 18.

Table 16 Operating hours at sea on free-piston gas turbines

Vessel	Country	Gasifier manufacturer		Plant rating[c]	Application[a]	Hours operation	Operation up to[d]	Vessel miles[b]
Cantenac	France	SIGMA	GS-34	2000	MP	13000	12/57	145000 [66]
Merignac	France	SIGMA	GS-34	2000	MP	13000	12/57	120000 [66]
William Patterson	U. S.	General Motors	GM-14	6000	MP	1320	3/58	18750
Minesweeper (21 ships)	France	SIGMA	GS-34	1800	NP	40000	12/56	
Bethsabee	France	SIGMA	GS-14	600 kw	MP	6500	6/57	
					Total	73820		518000 (estimated)

[a] P = propulsion, G = generator, M = merchant, N = naval.
[b] Distance vessel traveled with free-piston gas-turbine-propulsion in use.
[c] Rating of plant, not necessarily one gasifier, in shp unless otherwise stated.
[d] Approximate date.

Table 17 Nonavailability of the *Auris* 1200-shp gas turbine at sea [54]

	Date	Duration	Cause of stoppage	Steps taken to avoid repetition
1	January 14, 1952	6 hr	Failure to start, electrical fault in igniter power pack	New turbine has low-voltage carbon-rod igniters
2	January 18, 1952	15 hr	Joint blown in cooling-air pipe to expansion bellows	No recurrence after remaking joint
3	February 6, 1952	54 hr	Lubricating oil fire at h-p turbine, forward bearing	Shield fitted to prevent oil reaching hot surfaces
4	March 17, 1952	1¾ hr	Fire in heat exchanger	Steam sootblowers fitted
5	May 8, 1952	38 hr	Starting-motor gearing damaged	No recurrence after repair
6	July 17, 1952	100 hr	H-p turbine choked by ash deposits	Development of improved methods of cleaning turbine in service
7	February 24, 1953	7½ hr	L-p turbine seized	Design of sliding feet of pedestal modified
8	April 16, 1953	12¼ hr	Defect in alternator	No recurrence
9	July 8, 1953	9 days	L-p turbine bypass valve jammed	No recurrence after repair; new turbine has no bypass valve
10	September 6, 1953	4½ hr	Lubricating oil fire at h-p turbine, forward bearing	See 3, new turbine has improved bearing-sealing arrangement
11	May 25, 1954	9 days	Starter motor burnt out	This motor was rather highly rated; steam turbine for starting new set
12	September 18, 1954	73 days	Compressor thrust-bearing failure	Thrust collar of stainless iron replaced by nitrided carbon steel
13	September 10, 1955	2 hr	Electrical fault in trip circuit	No recurrence
14	September 15, 1955	9½ hr	Electrical fault in engine room, causing blackout	Improved arrangements for dealing with this type of emergency
15	October 3, 1955	13 hr	Igniter trouble and starting difficulties due to low turbine and compressor efficiencies	See 1, high compressor and turbine efficiencies can now be continuously maintained

Table 18 Fuel used in part of *Auris* trials [54]

Fuel	F 29
Viscosity, sec Redwood	
at 100 F	1628
Sp gr	0.957
Sulfur	2.5
Ash	0.077
Ash analysis,	
Vanadium	64
Ni as NiO	4.8
Na as Na_2O	7.8
Ca as CaO	...
Fe as Fe_2O_3	2.4
Sulfate as SO_3	19

Table 19 Typical analysis of Bunker-C fuel used in *John Sergeant* [55]

Gravity, deg API	18.9
Viscosity, SSF/122 F	88
SSU/210 F	107
Sulfur, weight per cent	1.55
Ash, weight per cent	0.023
Metals, ppm in oil	
Sodium	15
Vanadium	58
Nickel	19
Iron	9
Calcium	5
Magnesium	2
Lead	1

(*g*) *Fuel Additives.* Extensive investigations were conducted to develop methods of preventing blade deposition and corrosion, particularly deposits which tended to increase corrosion. Work was done to produce a fuel additive that would combine with the ash so the end product would have a melting point higher than the ash or the additive and would not deposit on the blades. Fuel additives investigated included ethyl silicate and aluminum silicate. Other additives did not give encouraging results. The new plant has provision for addition of insoluble material such as aluminum silicate as well as oil and water-soluble materials.

(*h*) *Airborne Additives.* Mechanical scouring of turbine blades was investigated. Materials including silica (80 and 200 mesh), kaolin, and crushed walnut shells were injected into the turbine-inlet ducts. Walnut shells (20–40 mesh) proved as effective as the silica. Cost of walnut-shell treatment was less than 0.5 per cent of fuel cost. The new plant will have provisions for injection of airborne particles into the turbine inlet to clean blades by mechanical scrubbing in addition to fuel-additive means.

John Sergeant [*55*]. This ship has, as of April, 1958, completed 4723-hr operation at sea. During this period the ship traveled some 62,000 miles. The power output was over 6000 shp with the exception of instances when reduced power was ordered by the bridge. The turbine plant was operated in an exceptionally fine manner. Troubles with the turbine and compressor have been nonexistent. It is significant that the over-all specific fuel consumption of the ship at design point has been about 0.530 lb per shp-hr.

(*a*) *Fuel Oil.* Both the maiden voyage and the second voyage were made on diesel fuel in order to give the engine-room personnel an opportunity to become fully acquainted with the plant before starting fuel treatments. On voyage no. 3, Bunker-C fuel was used for about 53 per cent of the time. At the end of voyage no. 6, some 2032 hr had been on Bunker C and 2002 hr on diesel fuel. The Bunker C specified in Table 19 is now being used.

A fuel analysis is required in order to determine physical and chemical properties prior to treating. A spectrographic method was initially employed. This required considerable time, about four hours, and a skilled operator. A new "rapid-field-analysis technique" is now in use which encompasses a quantitative analysis for vanadium and sodium—colorimetric and flame-spectrophotometric techniques respectively. An analysis is made in about one hour by a member of the engineering force.

(*b*) *Fuel Washing.* During voyages 3 through 6 specific gravity of the Bunker C was sufficiently low (0.940–0.944) that it was satisfactory to use fresh water alone, without additives, for proper separation of water-soluble materials in the fuel such as sodium and calcium. Water-wash consumption when at 6100 shp averages about 30 gal per hr. The centrifuges in this wash system have not had a forced shutdown during any of the Atlantic crossings. A European fuel of Mideast origin was not successfully washed on the westward passage of a recent voyage. This oil had an 0.976 specific gravity, 90 ppm Va, and 40 ppm Na. Tests are now being conducted to develop wash rates and emulsion breaker necessary for this fuel.

(*c*) *Fuel Additives.* The desired ratio of magnesium to vanadium in the fuel is controlled by addition of proper amounts of sulfate to the fuel. An aqueous solution of magnesium sulfate is metered into the Bunker C fuel ahead of the gas-turbine fuel pump. The additive system is automatic and operates only when the plant is burning Bunker C.

(*d*) *Inspection of Turbine.* Following voyage no. 4, after 2880-hr operation, about 900 hr on Bunker C, the turbine was given an inspection. Combustion piping had a fine powder deposit, yellow in color, of magnesium sulfate varying in depth to $^1/_4$ in. max. The combustion-chamber liners showed no indication of cracks. First-stage nozzles had some deposit which was 90 per cent magnesium sulfate on the leading edge. A dye check of first-stage nozzles revealed no cracks. A white-colored film of powdery deposit was found on the gas-side headers of both regenerators and the turbine-exhaust casing and ducts. Steam lancing of the gas side of the regenerator is not required more often than once in six months. The compressor blades and stator vanes had a light deposit of loose dirt with a white crystalline material at the end of a 10-day sea voyage. Analysis showed this to be 66 per cent sodium chloride and 11 per cent iron oxide.

(*e*) *Fouling.* There was a marked drop in over-all efficiency due to compressor, turbine, and regenerator fouling. Data show that regenerator fouling appears to level off after 10 days continuous running with a 10.5-per-cent decrease in regenerator efficiency. An increase of 12.5 per cent in the over-all fuel rate was found in a 240-hr Atlantic crossing. Compressor efficiency dropped 5.3 per cent, resulting in an increase of 9.5 per cent in fuel rate. The turbine and regenerator fouling account for the additional 3.5-per-cent change in fuel rate.

(*f*) *Compressor and Turbine Cleaning.* The method developed by the ship for removal of salt from the compressor and magnesium sulfate from the turbine is as follows [55]:

1 Cool unit down for 15 min at end of trip.
2 Refire for several minutes to blow out material broken free by step 1.
3 Shut down for 10 min.
4 Crank gas generator to 1400 rpm.
5 Check compressor drain water for color and taste. Repeat step 4 if water is not clear and free of salt.
6 Refire for 10 min to dry plant.
7 Shut down completely.

This cleaning procedure has proved effective in restoring normal efficiency of the plant at the end of each long passage.

40-Ft Personnel Boat [*56*]. The U. S. Navy installation of a 500-hp gas turbine in this boat was made to determine the advantages of the engine in a high-speed craft. This propulsion plant was described in detail earlier. Performance of the engine and reverse gear was outstanding. After 290 hr of running time a defective O-ring seal and an overheated circuit breaker were the only engine-part failures. A total of 800 hr of sea operation

was accumulated on this engine up to April, 1958. It is estimated that the boat traveled 16,000 miles with 1225 engine starts during the gas-turbine evaluation, mainly in open-sea operation at 97 per cent of rated speed. Engine maintenance included routine inspection of lines, filters, cleaning nozzles, combustor, and compressor at the end of each 100 hr.

The original power plant was a 225-hp diesel which gave the boat a speed of 12 knots. The 500-hp Solar T-522 gas turbine increased the speed to about 25 knots. Ahead reach from full speed of 25 knots to a dead stop was 90 ft. The turbine was capable of very rapid starting. In less than 40 sec from the time the starter button was pressed the boat was traveling at more than 20 knots.

LCPL (40-Ft Boat). The successful operating experience with a gas-turbine propulsion plant in a 40-ft Personnel Boat led to the decision to build a prototype LCPL. One boat of a new building program was equipped with a 500-hp gas turbine, Fig. 20, the others were diesel driven. Characteristics of the gas-turbine and diesel boats are included in Table 10. The gas-turbine boat has been operated by naval personnel for 275 hr with 1474 starts up to April, 1958. Performance has been good. The reverse reduction gear permits rapid stops. From about 23 knots full ahead the boat can be stopped dead in the water with an ahead reach of *150* ft within *9.5* sec. Ease of operation and simplicity were recently demonstrated when an inexperienced crew operated the boat after brief instruction.

HMS Grey Goose. Two Rolls-Royce 5400-hp gas turbines, Fig. 17, with intercooling, heat exchanger, and concentric turbines, straight compound cycle, were selected for this installation after careful analysis [10]. One reason for choosing this plant was its good specific fuel consumption at cruising. The engines in this ship accumulated a total of 2623 hr running at sea, divided 1344 hr for the port engine, and 1279 for the starboard. The design overhaul interval was 1000 hr, and they well exceeded this goal. The experience with this installation showed the engines and over-all plant performance to be excellent. However, the complexity and cost of manufacture make these engines less attractive than the simple cycle for high-speed craft.

HMS Bold Pioneer and Bold Pathfinder. Each of these fast patrol boats was propelled by two Metropolitan-Vickers G-2 Series I gas turbines, and a total engine time of 1664 hr was accumulated at sea. A number of development and operational troubles were experienced during initial trials. Air starting has been quick and reliable, idling speed was reached in about 30 sec, and full-power conditions were obtained in 12 sec after idling. Water-washing of the compressor was necessary to remove salt deposits and restore engine performance. Engines were usually washed each day by injecting about 10 gal of distilled water into the compressor at 5000 rpm, over a 3-min period. After high-power runs the gas generators were usually operated for 5 to 15 min at 4750 rpm to cool bearings to 194 F prior to shutting down. Installation of thermocouples in t.. bine exhaust rather than at the inlet is desirable because of greater reliability [10].

2. Auxiliary Gas Turbines. The auxiliary gas turbine has become well established in both merchant and naval marine use. The number of engines and vessels is predominantly naval. About 354 engines and 206 vessels are involved. Merchant applications include some 13 engines and 13 vessels, Table 1. Auxiliary sets have accumulated about 71,000 hr at sea, Table 22.

750-kw Generator. A Ruston Hornsby Mark-TA gas-turbine-driven generator, Fig. 32, has been in operation on *HMS Cumberland*, a "County-class" cruiser. This set supplies 440-volt, 60-cycle, 3-phase power. Control requirements were: Speed change not to exceed 2.5 per cent with load change of 25 per cent of rated full load; momentary voltage change not to exceed 15 per cent and to be restored to and kept within ±1 per cent of set voltage within 0.5 sec from initial change [57]. The set has been in operation for a number of months. As of December, 1957, it had accumulated a total of 1400 hours running time on the ship.

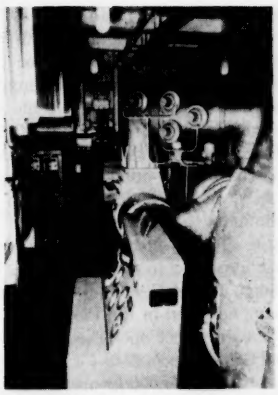


Fig. 32 Ruston Hornsby Mark T-A gas turbine driving a 750-kw generator set on *HMS Cumberland* (Courtesy British Admiralty)

Fig. 33 Cracks in the inner and outer rings of the upstream side of the first-stage turbine nozzle of a 500-hp gas turbine after 1500-hr shipboard test (Courtesy Solar Aircraft Company)

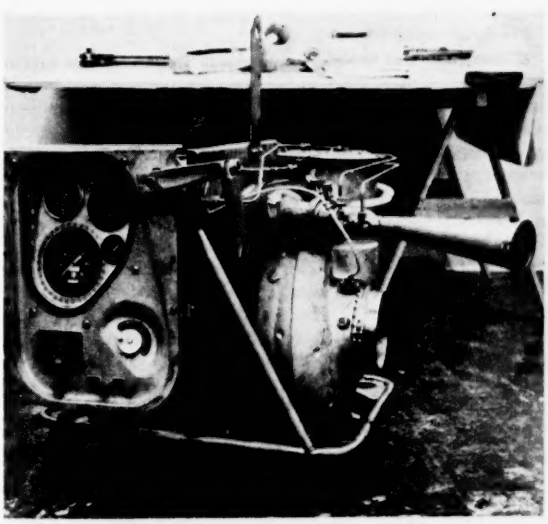


Fig. 34 Gas-turbine air-bleed unit (Courtesy Solar Aircraft Company)

300-kw Generator. This Solar gas-turbine-driven set, Fig. 4, was installed on the *USS Gyatt* in November, 1956. The total hours, including test-stand and ship operation, are 258 with over 700 starts. Since installation it has been operated a total of 80 hr and started 136 times. Immediately following the ship installation the starter failed to function properly and engine combustion was not maintained. Modifications to the starter

and turbine-fuel control cleared up all troubles [8]. The set is operating with excellent reliability and meets the 10-sec start requirement consistently. Shipboard tests of two similar gas turbines were conducted for a total engine running time of 3000 hr [14]. During these tests, parts were replaced as shown in Table 20. After completion of the shipboard tests, the engines were overhauled and compressor-cleaning techniques evaluated. Cracks developed in turbine-nozzle ring assemblies, Fig. 33. Based on these tests, design changes have been incorporated in new engines to eliminate the part failures.

200-kw Generator. The Boeing 502-6 gas turbines driving an auxiliary generator have been in service on U. S. Naval vessels for several years. Some 184 of the 502-6 engines are in this type service. They have operated in the Fleet in excess of 65,000 hr [58]. During the early Fleet operations turbine-blade and accessory-drive-gear failures accounted for the greatest down time [17]. This was in the first 20,000-hr operation. Design changes incorporated in the engines subsequent to these failures have greatly improved engine reliability. An accelerated test [59] on two of these engines with improved components was conducted. The test was initially set for 1000 hr. However, inspections showed the engines to be in such excellent condition that both were operated for a total of 5000 engine-hours. The test was concluded primarily because of a severe crack that developed in the disk of the power turbine on one engine. It is significant that many of the major components on these two engines had 600-hr operation prior to starting this test. A service record of these components is shown in Table 21. During the last 1000 hr of test the engines had a 99.4 per cent reliability.

The auxiliary application of the gas turbine as a drive for fire pumps is increasing. Units in this service are proving to be reliable and simple to operate. They have been exceptionally good under cold-weather conditions. Other applications include air supply, Fig. 34, fog generators, and deicers. Operation of the engines in this intermittent service does not result in a rapid build-up of engine-hours.

B Free-Piston Gas Turbines. As shown in Table 1, the free-piston gas-turbine plant is most widely used for propulsion. At this time 36 ships have, or are being fitted with, free-piston propulsion plants. The 600-kw set in the French tanker *Bethsabee,* is the only generator set with any appreciable hours at sea. Propulsion units have accumulated over 67,000 hr at sea. Operation and maintenance experience on the free-piston plants have been summarized in earlier papers [22], [30], and [60]. In the French coasters *Cantenac* and *Merignac* availability has been good. The *Cantenac* was out of service 8 days in 2½ yr of operation. The *Merignac* was down 4 days in 2 yr. Ring life did not exceed 500 hr in these vessels during early operation. With improvements in ring material, design, and lubricant, operating periods without ring inspection, greater than 2500 hr were achieved. Oil consumption was high at first due to leakage at seals on the synchronizing rods and piston cooling-oil tubes. Following improvements of surface finish on the rods, packing-ring material and design, and oil-outlet pipe the consumption of circulating oil is about 0.00028 lb per hp-hr. Engine-case fires resulted from the oil leaks which caused carbonaceous deposits in the scavenge casing. The fires occurred after several hundred hours operation. These fires were finally prevented by: Lowering engine-case temperatures (below 446 F) with intensive cooling; improving compressor efficiency; elimination of hot points; selection of oil more compatible to the operating conditions; and elimination of oil leaks.

C Specific Areas of Operation. Once a new machine is developed to a point of reasonably competitive performance and reliability, it must be taken out of the laboratory and placed in operation. When the first gas-turbine ship, the Royal Navy's MGB 2009 powered with a Metropolitan-Vickers G-1 2500-hp turbine went to sea in July, 1947, the "new machine" was still

Table 20 Replacements made during 1500-hour shipboard test of two Solar T-520J gas turbines driving electric generators [14]

Part replaced	Number
Turbine air seal	12[a]
Combustor liner	9
Fuel pressure gage transmitter	6
Spark plug	3
Speed-sequencing switch	3[a]
Starter	3[a]
Lubrication oil pump	3
Acceleration limiter	2
Fuel nozzle	2
Fuel accumulator	2[a]
Fuel system piping	2 sets[a]
Oil pressure switch	1[a]
Fuel-boost pump	1[a]
Control relay	1[a]

[a] Failure caused forced shutdown.

Table 21 Major turbine-component record during 2500-hr test on two Boeing 502-6 engines driving a generator [59]

Turbine	Component	Hours
1	First-stage turbine	2500
2	First-stage turbine	2500
1	Second-stage turbine	3100
2	Second-stage turbine	2600
1	Compressor impeller	2600
2	Compressor impeller	2100[a]
1	Compressor case	2500
2	Compressor case	3100
1	Accessory drive unit	3100
1	Lube-oil pump	3100
2	Lube-oil pump	3100
1	Reduction gear	3100
2	Reduction gear	2600
1	Nozzle box, first-stage	2500
2	Nozzle box, first-stage	2600
1	Nozzle duct, interstage-bleed	3100
2	Nozzle duct, interstage-bleed	3100
1	Exhaust-collector assembly	3100
2	Exhaust-collector assembly	3100
1	Exhaust system and components	3100
2	Exhaust system and components	3100
	Combining gear	3100

[a] Impeller replaced at 2100 hr to eliminate suspected cause of vibration.

Table 22 Summary of operating hours at sea on gas turbines and free-piston gas turbines

Engine	Service		Sea hours
Gas turbine	Propulsion		
	Merchant		24000
	Naval		8000
		Total	32000
	Auxiliary		
	Merchant		...
	Naval		71000
		Total	71000
		Gas turbine total	103000
Free-piston gas turbine	Propulsion		
	Merchant		27000
	Naval		40000
		Total	67000
	Auxiliary		
	Merchant		6000
	Naval		...
		Total	6000
		Free-piston gas turbine total	73000
		Grand total	176000

in the laboratory; a floating one. In the years immediately following this major step in the marine-propulsion field other floating laboratories appeared. Then in 1950, small engines, the Boeing 502-6, were placed in marine service on a "use" basis and not as experimental plants. The increase from 2500 hp in 1947 to more than 310,000 hp at the present is actually a tribute to the foresight of the designers, builders, and purchasers of the gas turbine. With the knowledge gained through operation at sea, improved engines are appearing in more areas. Searching questions have been answered through this sea experience. A number of the important areas will be discussed:

Reliability and Availability. The per cent reliability has been defined as [61]

$$\text{Per cent reliability} = \frac{(\text{installed hr} - \text{forced-outage hr}) \times 100}{\text{installed hr}}$$

These are of paramount importance. It has been demonstrated that gas turbines are satisfactory to propel rather expensive aircraft. For example, eight Pratt and Whitney J-57 gas turbines power the $8-million B-52 airplane. Experience on the *John Sergeant*, a single-screw vessel, has shown reliability equal to a steam turbine or diesel during the operation to date. Industrial turbines similar to the *John Sergeant* have demonstrated a reliability greater than 99 per cent. These industrial units also have a use factor greater than 99 per cent defined as

$$\text{Per cent use factor} = \frac{\text{total fired hr} \times 100}{\text{total installed hr}}$$

The availability of these industrial units has been over 99 per cent [6], defined as [61]

$$\text{Per cent availability} = \frac{\text{installed hr} - (\text{forced outage} + \text{maintenance}) \text{ hr} \times 100}{\text{installed hr}}$$

Reliability and availability of the 300-kw generator set in the *USS Gyatt* has been 100 per cent since the initial starting problems were resolved. The set has 275 hr total operation, including test stand and ship. During the early months of operation, trouble was experienced with the starter and fuel scheduling. After solving these two problems, the engine has had an exceptionally fine record. From March, 1957, to April, 1958, 136 starts were made with only one start failure which was caused by a low battery. The engine was started immediately after this failure. The set tripped out once under load, cause undetermined, since March, 1957. These two cases of malfunction probably resulted in no more than 5 min down time. Thus reliability and availability have been practically 100 per cent.

Operating Cost. The operating cost of lightweight and heavy-duty-type gas turbines is lower than competitive prime movers. The Dart engines in the Viscount airplanes in the United States have established an overhaul cost of $3.51 per engine-operating hr [20]. Reliability of this engine has resulted in a reduction of spare engines initially purchased. Overhaul costs of 5 to 10 per cent of the original gas-turbine cost are expected. This compares to cost of a major overhaul of 30 to 40 per cent of new engine cost for reciprocating engines [14]. Comparison of maintenance costs of three types of prime movers is as follows [13]:

Prime mover	Cost of maintenance in per cent fuel cost
Gas turbine	1.4
Diesel	46
Steam	10–20

A further comparison of maintenance costs of various prime movers is [12]:

Gas turbine on *Auris*, propulsion	0.06 cent per kwhr
Brown-Boveri gas turbine (based on service life of 100,000 hr)	0.023 cent per kwhr
Diesels (British Diesel Engine Users Association)	0.30 cent per kwhr
Steam turbines, without boilers (British Electricity Authority)	0.049 cent per kwhr

A recent comparison between gas turbines and piston engines operating under the same conditions shows operating and maintenance costs [40]:

	Cost, pence per kwhr	
	Operation	Operation and maintenance
Gas turbine	0.196	0.338
Piston engine	0.381	0.643

Operating Personnel Required. The number of personnel required to operate a gas-turbine plant is generally 15–30 per cent less than for diesel or steam plants. The *John Sergeant* requires 14 per cent less personnel than a comparable diesel ship, the *Thomas Nelson*, and 25 per cent less than the steam-turbine ship, the *Benjamin Chew*. The requirement for less personnel is due to simplicity of the gas-turbine plant. This is a result of fewer supporting auxiliaries, more automatic controls, basic simplicity of gas turbines, and the lack of inherent plant complexities.

Noise. Silencing of marine gas turbines has proved to be a relatively minor problem. In machinery areas that are manned when turbines are in operation, adequate silencing of inlet and exhaust can be realized. Use of splitters and sound-absorbing materials in ducts is effective. During maximum-power trials on the *John Sergeant* [21] noise levels were recorded. Readings in decibels were:

Gas-turbine unit	96–98
Main reduction gear, aft end	102
Main reduction gear, forward end	105–107
Air intake above bridge deck	102
Motor-driven air compressor	106
Diesel-generator set	116

Turbine installations in boats have been acceptable after proper silencing treatment.

Compartment Ventilation. Engine enclosures and compartments in many installations are supplied with cooling and ventilating air through eductor action of stack gases. A boat-propulsion enclosure cooled by the stack eductor is shown in Fig. 35. An eductor system in a 40-ft boat powered with a 500-hp gas turbine had a cooling-air flow of 800 to 1000 cfm. This was accomplished without any increase in exhaust velocity and therefore no loss in

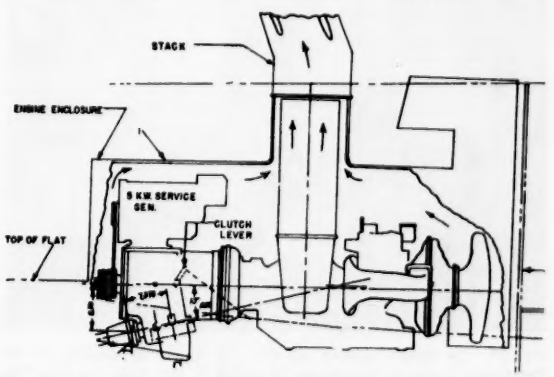


Fig. 35 Compartment ventilation, by means of stack eductor, in a boat-propulsion package

propulsion power resulted [35]. Eductor design methods are discussed and illustrated in references [62] and [63].

Compressor Cleaning. During sea operations salt is carried into the compressor along with combustion air. Deposition of salt on blades and stator vanes results in a decrease in engine performance. Test on the Bristol Proteus engine, where large quantities of sea water were injected into the air inlet, showed a power drop from 2825 to 2105 shp, a loss of 720 shp, after 125 hr running [9]. This was due to salt deposition on the compressor. An injection of two gallons of fresh water gave a full power recovery. In the *Bold Pioneer*, with two 4500-hp turbines, compressor washing with 10 gal of fresh water has proved adequate [10]. Where compressor fouling is by oil deposition, injection of solids in the air is more effective than water-washing [14] on a 500-hp engine. Washing is usually most effective at reduced speeds. In the *Auris* 1200-hp gas-turbine compressor cleaning at full power was accomplished by introducing 10 lb of flour-mill by-products into the inlet in a period of 1 min [54].

Ducting. Proper sizing of inlet and exhaust ducts is essential to

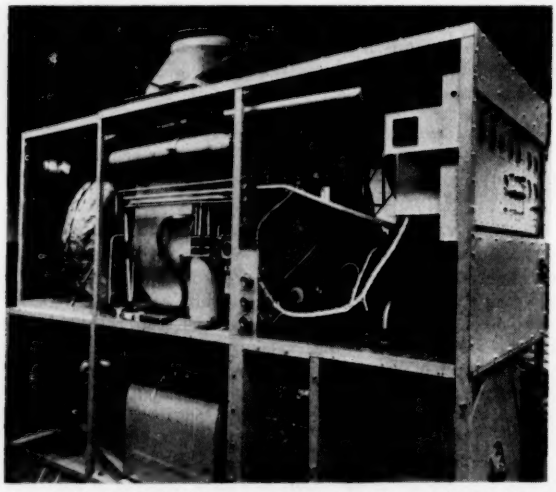


Fig. 36 Piggy-back 300-kw generator set driven by a Solar T-520 gas turbine (Courtesy Solar Aircraft Company)

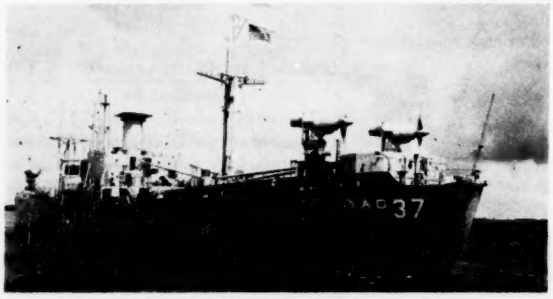


Fig. 37 YAG-37 Liberty hull propelled with four Pratt and Whitney T-34 gas turbines driving air propellers (Courtesy U. S. Navy)

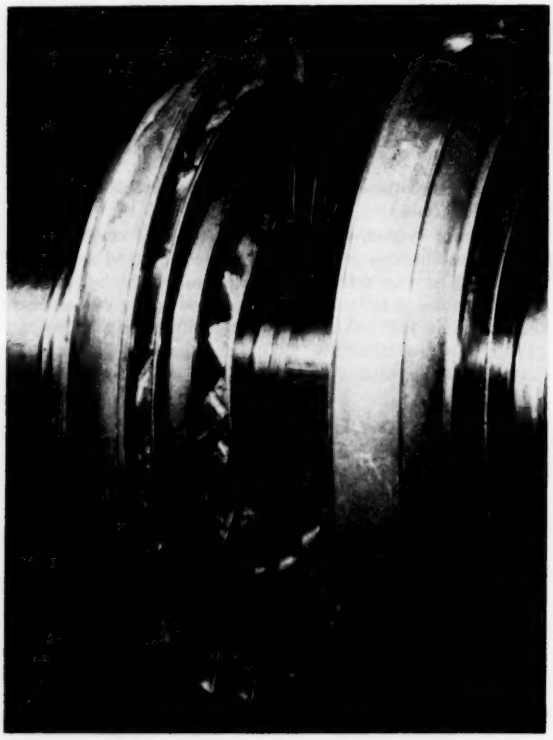


Fig. 38 The astern rotating element, not finally assembled, of the Pametrada hydraulic reverse gear installed in the *Auris* (Courtesy The Parsons and Marine Engineering Turbine Research and Development Association)

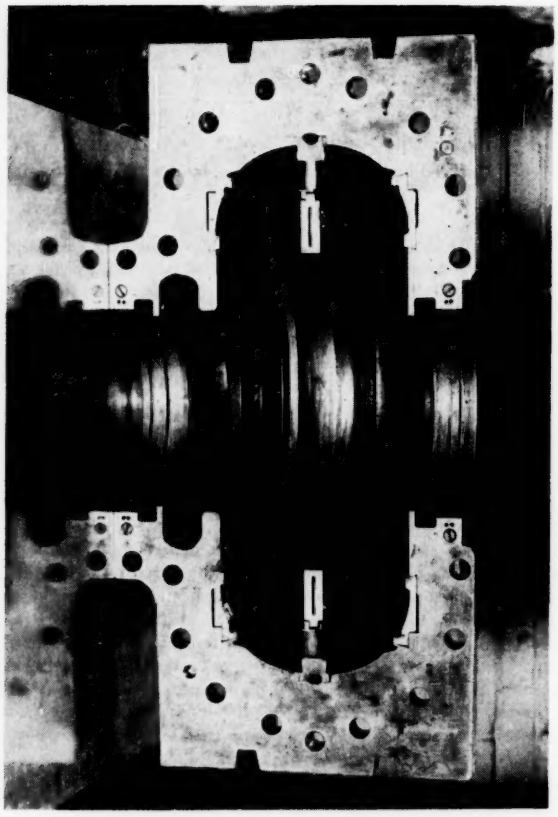


Fig. 39 The astern fixed casing, without rotating elements assembled, of the Pametrada hydraulic reverse gear installed in the *Auris* (Courtesy The Parsons and Marine Engineering Turbine Research and Development Association)

minimize pressure losses and thus obtain maximum power output. Selecting the most advantageous location for the turbine in a marine installation can go a long way toward reducing duct losses. The cost of capacity lost in a 7100-hp plant by duct losses has been estimated for a power station: With fuel cost of $0.20 per million Btu, load factor 60 per cent, 20 per cent fixed charges, and an installed cost of $200 per kw. These estimates for a 5 in. H_2O pressure drop, assuming power would have to be purchased to make up the loss, are [64]:

Pressure drop,	Approximate cost in dollars per yr		
in. H_2O	Inlet	Exhaust	Total
5	10,000	6,000	16,000
10	20,000	12,000	32,000

Astern Power. Free-piston propulsion plants, including 21 French minesweepers and the *William Patterson*, are in operation with astern stages in the turbine for reversing the propeller. While in ahead operation the astern stages cause some loss due to windage. These losses are reduced to less than 2 per cent of rated power by means of a mask fitted near the astern blades [30]. Care must be exercised to prevent overheating of ahead stages during prolonged periods of astern operation at high power. Gas-turbine propulsion plants are in operation with controllable-pitch propellers and mechanical reverse gears. The plants with mechanical reverse gears have been limited to about 500 shp until the recent launching of the *Brave Borderer* equipped with three 3500-shp Proteus gas turbines [34]. Experience with the mechanical reverse gears has shown reliability and maneuverability to be entirely satisfactory. Controllable-pitch propellers have found wide acceptance in the field of aviation and they are now being used in some large ships. The *John Sergeant* installation has illustrated the rapid-maneuvering capability of a ship equipped with a controllable propeller. . Trials showed that from an ahead speed of 17 knots to a crash astern the ship was dead in the water in 2 min 52 sec with an ahead reach of 757 yd measured by Raydist. The time for a C2 to be dead in the water from full power ahead to a crash astern is about 4 min [21]. Experience with the hydraulic mechanical reverse gear of Pametrada design, now installed in the 5500-shp *Auris* will be of considerable interest when it becomes available.

Starting. Methods of starting gas turbines used in marine service vary considerably and depend to a large extent on demands set by operating conditions. The following means of starting are now being used: (*a*) Combustion-air starter; (*b*) electric motor; (*c*) steam turbine; (*d*) diesel; (*e*) auxiliary gas turbine; (*f*) hand cranking; (*g*) air motor; (*h*) hydraulic motor.

Residual Fuel. The free-piston plants are burning residual fuels. However, the gas-turbine installations operating on residual fuels are relatively few. The *Auris*, 1200-hp turbine, burned residual for 6649 hr out of a total operating time of 20,510 hr. The *John Sergeant* has operated some 4723 hr at sea, through April, 1958. A total of about 2650 hr were on Bunker-C fuel. The turbine-inlet temperatures were 1108 F and 1450 F on the *Auris* and *John Sergeant*, respectively. Both installations, while operating at widely differing turbine-inlet temperatures, have found it necessary to devise means of cleaning combustors, nozzles, turbine blades, and regenerator surfaces [54, 55].

Overhaul Interval. With respect to routine inspection, maintenance, and overhaul the gas turbine is similar to other prime movers. Inspections, maintenance, and overhauls must be conducted at established intervals. The frequency varies with the type engine and duty. Overhaul intervals for a number of gas turbines are shown in Table 8. It is of interest to note that the intervals vary from 200 to 100,000 hr.

Regeneration. This means of improving plant efficiency has been utilized in the *Auris* and *John Sergeant*. Operational difficulties on the *Auris* consisted of one split tube after 2-yr service. At the end of five years there were no more defective tubes nor any sign of corrosion.

Conclusions

As a result of the relatively large increase in marine applications during the past five years, it is concluded that the gas turbine is now an accepted prime mover for ships and boats. The increase has been due to attainment of satisfactory levels of: (*a*) Reliability; (*b*) low maintenance; (*c*) low operating cost; (*d*) availability; (*e*) economy; (*f*) low number of operating personnel; (*g*) maneuverability; (*h*) minimum weight; (*i*) small space; (*j*) rapid starting.

With the development of new-type ships such as 1000-ton hydrofoils, where greatly increased emphasis will be placed on higher powers in small volume, the gas turbine will find increased use. As a result of the continued research-and-development programs of government and industry, turbine over-all efficiencies will be greatly improved. It is estimated that, with present materials and techniques, relatively small gas turbines can be built in the near future that will have lower specific fuel consumption than spark-ignition reciprocating engines [65]. With increases in cycle temperatures that will soon be feasible, even greater improvements will be made in reducing fuel consumption. Propulsion gas turbines in large powers are now realities in merchant service, and a number of naval plants are building. These include both base-load and boost-power units.

Powers demanded for high-speed ships and boats will of necessity require gas turbines. Other prime movers cannot meet the specific power and volume specifications.

Auxiliary applications, both merchant and naval, are moving at encouraging rates. Uses now include: Generators, boiler superchargers, fire pumps, portable generators, fog generators, air-supply units, and aircraft starters. New fields are being explored and present applications are under expansion.

The gas turbine is not expected to replace the steam turbine and diesel in all types of marine service. However, the gas tur-

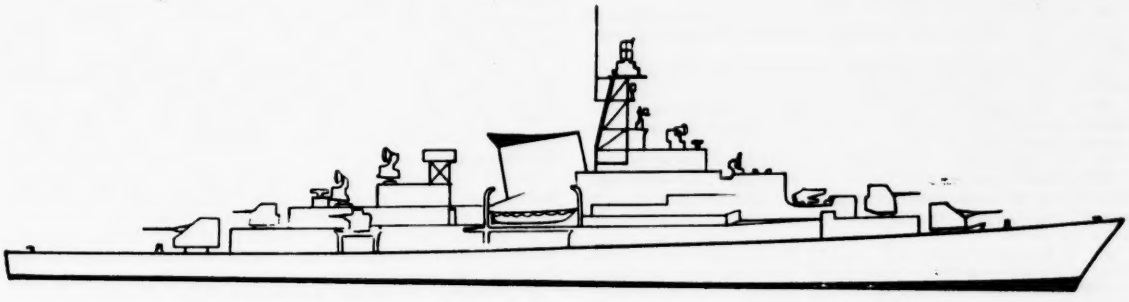


Fig. 40 Sketch of German naval escort ship now under construction (Courtesy *The Motor Ship*)

bine does many jobs more economically than either steam or diesel. It is predicted that the gas turbine will be used in the marine field at a greatly accelerated rate.

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DISCUSSION

W. M. M. Fowden, Jr.[2]

It is obvious that the gas-turbine engine has characteristics which are highly desirable, indeed necessary, for ships and boats. These characteristics have been thoroughly discussed by the authors in the early pages of their paper. However, in spite of these many desirable characteristics, the use of gas-turbine engines in ships has not been as large as we would expect. The authors have noted this fact briefly on page 326.

Possibly the application of the gas turbine has been shadowed by the use of nuclear power in ships. However, high initial cost has probably been the most important factor retarding the application in recent years. The requirement for low cost is mentioned on page 312 and again on page 328. Many applications of the gas turbine have been made in spite of initial cost where the special character of these machines has warranted. Other installations have been made "to gain operating experience" or for similar reasons. The authors mention low initial cost very briefly on page 314. It would be helpful if this portion could be amplified so that we can see the trend toward the low initial cost requirement. Can the authors provide a dollar per hp versus time curve for large and small gas turbines which will show the sharp drop in initial cost?

Recent news releases by the U. S. Navy have mentioned the technical breakthrough in ship propulsion attributed to the "supercavitating" propeller. Some of this information has linked the gas-turbine engine with the advantageous application of this propeller. It would be especially interesting to hear from the authors how this matching might influence the use of gas-turbine engines in ships and boats. What significant improvements in the propulsion system will be made through reduction in weight or maneuverability, for instance?

[2] Commander, U. S. Naval Postgraduate School, Monterey, Calif. Mem. ASME.

The PT-812 installation and the U. S. Navy Cosag development have been mentioned for several years in papers before this and other societies. Each of these projects would seem to be worthy of more detailed discussion than can be made in a paper such as this one. Would it be possible for the authors to instigate a comprehensive presentation of these power plants in the near future?

The authors are to be congratulated on their treatment of marine gas-turbine developments over the past five years. This paper with the others in this series should—when the facts are digested—give us an idea of where we are going as well as where we have been.

A. A. C. Gentry[3]

The authors have covered a very wide field in this paper which gives an interesting survey of the increase in gas-turbine applications over the years since the first installation in MGB 2009.

The advantages of the gas turbine in weight and space can be utilized to various degrees either for short life and high power/weight applications or for long-life applications as quoted by the authors. It is true to say and, in fact, is illustrated in this paper, that the former applications are, in general, of a specialized nature, e.g., Naval fast patrol boats, high-speed personnel boats.

It is in the field of long-life applications that gas turbines must achieve marked advantages in simplicity of operation and reduced maintenance as well as the best possible fuel economy and a competitive manufacturing cost if continued consideration for more widespread marine applications is to be maintained.

The initial experience gained by the Royal Navy has led to the adoption of gas turbines working generally on simple cycles in those applications where gas turbines show to advantage as illustrated in this paper. The most complicated cycle now in use in the Royal Navy is that employed in the W. H. Allen 500-kw generator—a two-shaft machine with intercooler. Effort, however, is now being concentrated in achieving reliability and simplicity of operation with reduced maintenance on present designs rather than to try to achieve optimum advances in economy by the development and reintroduction of more complicated cycles such as RM 60.

The installation of the W. H. Allen generating set in *HMS Llandaff* referred to in the text has emphasized the problems of recirculation of exhaust gases, also mentioned in this paper, and of fouling from ventilation exhausts. The seriousness of the effects of compressor fouling from such sources must of necessity vary with the design of the gas-turbine unit, but some degree of fouling will, of course, be present in all designs. It has been necessary, therefore, as a result of this latest experience, to be particularly careful in the design of installations to position compressor inlets, turbine exhausts, and ventilation exhausts from such compartments as engine rooms and galleys as favorably as possible to maintain performance and to keep maintenance to a minimum. The W. H. Allen set has recently been accepted for ships' use as base-load generator and it is hoped that some valuable experience of this set in this role will soon be forthcoming.

The Ruston T.A. alternator fitted in *HMS Cumberland* has continued to perform satisfactorily and has been used to assess degree of fouling and compressor cleaning routines by means of solids, e.g., wheatings, ground nut shells. It must be mentioned that this set rated at 750 kw has actually run at loads not exceeding 250 kw for the majority of its installed time. Similar work on compressor cleaning is also being carried out on a Ruston T.A. gas turbine by the Naval Marine Wing of the National Gas Turbine Establishment.

The gas-turbine units for the Cosag propulsion plants in the

[3] Commander, Marine Engineering Division, Ship Department, Admiralty, Bath, London, England.

Royal Navy, referred to in the text, have been designed to achieve a relatively long-life gas turbine. The original pure "boost" conception has been modified to one of "dual machinery." This means that the design now allows for the use of the gas turbines alone for propulsion and maneuvering. To enable this to be carried out gas turbo-alternators will be installed for base-load requirements. The experience gained from these installations will therefore be of great importance in establishing the further possibilities and advantages of an all gas-turbine long-life propulsion system for Naval vessels, leading, possibly, to the authors' conclusions that gas turbines will be used in the marine field at a greatly accelerated rate.

It is suggested, however, that the authors' comments and conclusions regarding the use of higher cycle temperature do not mention the problems that may be met with in open-cycle gas turbines operating in a marine atmosphere. Experience in the Royal Navy to date indicates that a mean maximum cycle temperature of 1100 K should not be exceeded if corrosion problems associated with a marine atmosphere and sulfur bearing fuels are to be avoided.

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S. D. Hage[4]

Directing the evolution of a new technology in an efficient manner involves two very important ingredients, (*a*) a periodic review of fundamental principles involved, (*b*) a periodic collection of a mass of statistical data so progress can be actually measured.

The authors have taken advantage of their central location in the technology to provide industry with an excellent summary of the latter type. Industry should be very appreciative.

Everyone must know that the effort required for such a compilation is a large order. It is so large that personal dedication of an extreme degree is always required.

R. L. Jackson[5]

The authors must be commended for this fine work in reviewing the progress of marine gas turbines to date. The almost 70 references attest to the magnitude of their review effort. Of particular interest is the conclusion that gas turbines are now an accepted prime mover for ships and boats. That conclusion demonstrates the rapid progress which gas turbine applications have made in the past several years.

The applications which have been made demonstrate the extreme flexibility and adaptability of gas turbines to specific uses and requirements. Applications range from large ship propulsion to small fire pumps and fog generators. Sizes range from 50 to 7000 hp. Construction varies from very lightweight short-life machinery, to more heavily constructed, very long-life equipment. A variety of cycles is represented from simple cycle plants to those embodying both intercooling and regeneration. All of the higher efficiency plants, of course, employ at least regeneration. Specific fuel consumption has ranged from 0.5 to about 2 lb per shaft hp hr so that operating experience has been attained with a large number of different kinds of plants.

When gas turbines are considered as main propulsion plants, the problem of providing astern power is of great interest. The problem has been of particular importance for the larger installations perhaps not for technical reasons but for the costly development involved. The controllable pitch propeller has been demonstrated to offer a workable solution up to about 7000 hp and there are no apparent technical reasons why larger sizes are not equally feasible. There is much interest in the reversing hydraulic coupling installed in the Auris and it is hoped that the experience with this reversing gear will be reported soon. Radial inflow turbines with adjustable inlet vanes on the power turbine are a promising solution and the developments in this regard, which are being carried out by the Bureau of Ships, are welcome and it is hoped that the results of these developments can be reported soon. Although the progress report covers the *John Sergeant* experience it may be of interest to include some more recent information. The ship has now completed ten voyages and is now being operated by United States Lines in regular commercial service. The usual fuel consumption so far has been about 0.52 lb per shaft hp hr for the over-all ship with some voyages being as low as 0.49. With respect to plant fouling the regenerator is not being cleaned at all. The turbine is being washed at the end of each long voyage. On the first voyage considerable salt spray was induced in the compressor resulting in some fouling and depreciation in performance. More recently a hood has been fitted over the side opening air intakes so that air can only be drawn from aft thus reducing greatly the amount of salt spray induced. The worst fouling which has been encountered since then has resulted in a loss of compressor efficiency of less than $1^1/_2$ per cent and this was only during extremely dirty weather. The average decrease in compressor deficiency during the voyage is much less. The compressor is also washed with fresh water at the end of each voyage.

I think that we can look forward to continuing progress in the application of gas turbines for marine use and the continuing expansion in the number of gas turbines used at sea. The applications and experience reported in this progress report, and our own experience, demonstrate that gas turbines do offer installation and operating advantages in marine applications.

B. G. Markham[6]

The authors of this paper have produced a most thorough and painstaking survey of the present position which will serve as a valuable reference to all concerned with marine gas turbines.

They have completely succeeded in what they set out to do and it is appreciated that this could not have been done in a reasonable space had they included any considerable comments on the facts or expressions of their own opinions.

However, those of us on both sides of the Atlantic who have learned to value and respect the authors' views and judgments must feel a little disappointed that they have confined themselves so rigidly to statements of fact. I hope that this will be put right during the discussion and that we shall have the benefit of hearing the authors' present views.

I would particularly like to have the authors' opinions on the future of combined plants. It seems to me that for certain types of vessel a good case can be made for a gas turbine which is relatively low in power in relation to the diesel engine to boost up the speed. With this arrangement the diesel engine must have an overspeed rating to cover the increased rpm which will be obtained at a given throttle setting when the gas turbine is used.

At the other end of the scale there is a case with very fast vessels, which require only perhaps 5 to 10 per cent of their full power for maneuvering, for the fitting of a very low-powered diesel engine which may weigh little more than the reverse gear for a gas turbine. Such an engine would be fitted primarily for maneuvering but would also be available for cruising at low powers.

In between these extremes there are combinations in which the gas turbine and diesel engines are of approximately the same

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[5] Gas Turbine Department, General Electric Company, Schenectady, N. Y.

[6] Bristol Aero-Engines, Limited, Bristol, England.

power. Such an arrangement seems to combine the disadvantages rather than the advantages of the two types of engine. The design of a propeller for the diesel engine has to be a most difficult compromise as it must allow the diesel engine to produce useful power with the gas turbine running and yet must not give rise to excessive diesel-engine temperatures when the diesel engine is used by itself.

However, if a diesel engine can be fitted with an efficient controllable pitch propeller, which is not an easy thing on very high-speed craft, some of these difficulties are overcome.

In the U. S. Navy's PT 812, the diesel power is small but by no means negligible in comparison with the power of the gas turbines, and controllable pitch propellers are fitted. It would be interesting to know whether these propellers have enough range to allow the diesel engines to be used efficiently when the gas turbines are running or whether it is the practice to stop the diesel engines and to feather their propellers when the gas turbines are used.

It would also be interesting to have the authors' views on Cosag plants. The gas turbine must be fitted with a clutch for "Ahead" and in view of the very excellent maneuvering characteristics of a gas turbine, it must be tempting to consider the provision of an "Astern" clutch as well, which would allow the vessel to be maneuvered on the gas turbines alone and to be got under way, and perhaps taken to sea for short passages without getting up steam. One then begins to wonder whether one has a gas turbine boosted by steam or a steam plant with a gas-turbine boost.

Would the authors agree that the reasons for the existence of Cosag plants are due more to the inherent conservatism of marine engineers and the nonavailability of really high-powered gas turbines than to any inherent technical advantages in combined plants?

Paul A. Pitt[7]

It is clear from this comprehensive report that the gas turbine has made striking progress in marine applications during the past five years and in view of the progress being made in the art of design and technology, it is interesting to contemplate what the next five years may have in store for it.

The gas turbine is less than 20 years old and there are few examples in the history of engineering where in such a short period of time an utterly different machine has progressed so rapidly and has had such a wide influence on the concept of power as has the gas turbine.

Information and data assembled by the author are testimony to one of the most significant features of the gas turbine—versatility. It has no equal in its adaptability to perform many different functions. It is also unique in that its functional versatility is implemented by the arrangement of its components and/or the way in which these components are matched. This enables the gas turbine to deliver its output in several different forms:

1 *Shaft Power*
   (*a*) Variable speed, for propulsion of ships, land vehicles, and so forth
   (*b*) Constant speed, drive for electric generators, pumps, and compressors
2 *Gas Power*
   (*a*) Jet, for propulsion of aircraft
   (*b*) Heat, for boilers and chemical processes
3 *Compressed Air*
   (*a*) Air bleed, to supply compressed air

The report illustrates another aspect of the versatility of the gas turbine in the wide variety of component types with which the power plant can be built up to produce various characteristics. It may be made exceedingly simple, exemplified by the Solar Mars portable fire pump, or it may be made relatively complex to attain a high thermal efficiency and flexibility, as with the new British Thomson-Houston *Auris* turbine, or the Rolls Royce RM-60 turbine.

The table prepared by the authors on turbines available for marine service shows a range of power from 45 hp to 22,500 hp. Between these two power plants, there is a wide variety of design and power ratings. On the basis of available "know-how," practical limits on the upper and lower ranges of power have not yet been determined and it is likely that there will be much larger and much smaller power plants in existence in the future.

While it deals mainly with marine applications, the substance of the report leads to the conclusion that the gas turbine is finding its niche among prime movers and that it will continue to encroach upon the domain of the steam and reciprocating engines. Whatever success it achieves, whether it be in sea or land application, credit is due to the U. S. Navy, who have contributed so much to the backlog of experience with gas turbines through service application. The authors have played no small part in this pioneering effort.

John H. Strandell[8]

The writer desires to add a few words about a device that should always be installed with a ship's propulsion plant of the gas-turbine type. The reason is that no reversing device of any other kind can take its place and it is also a "time proven" piece of machinery produced by several companies the world over. The device here in consideration is the controllable pitch propeller. It not only reverses the thrust but also accomplishes this maneuver ten times faster as compared to reversing the rotation of a fixed pitch propeller, provided that reversing of both systems is conducted in the same manner. Both systems should be compared without introducing a second party executing the order from the bridge. In other words, both systems should use direct bridge control. Otherwise the difference will be much greater in favor of the controllable pitch propeller.

Many captains of large (7000 dw tons) ships equipped with twin controllable pitch propellers dock the ship without aid of tugs by using the controllable pitch features of the propellers. The controllable pitch features are also useful in accommodating to the power available due to variation in ambient temperature for open-cycle-type gas turbines, as operation of the *John Sergeant* has proven. The enormously improved crash-stop capabilities could act to materially reduce collision accidents. Sea and displacement conditions can also be adjusted for by utilizing the controllable pitch features. With the great activity that is now going on in the controllable pitch propeller field these claims cannot more convincingly be proven than to review all the long lists of controllable pitch propeller applications received from various controllable pitch manufacturers the writer has been in correspondence with for the last three quarters of a year.

The largest group of controllable pitch propeller installations is, of course, ships and boats with towing and maneuvering functions such as tugs, minesweepers, ice breakers, dredges, fireboats, trawlers, and fishing vessels of all types. Ships requiring considerable maneuverability are most often equipped with twin screw controllable pitch propellers such as ferries, speedboats, patrol boats, torpedo boats, and pilot boats. There is also a great number of single screw controllable pitch propeller installations in coastal and canal passenger and freight ships. An increasing number of controllable pitch propellers of large power (9,000 to 10,000 shp) is now being installed in dry cargo ships, freighters,

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[8] Propeller, Shafting, and Bearing Branch, Code 644, Bureau of Ships, Navy Department, Washington, D. C.

passenger ships, and tankers. Also large controllable pitch propeller installations can be found in German and British war vessels of escort-destroyer type.

The Navy Department, Bureau of Ships, has been endeavoring for more than 22 years to develop reliable controllable pitch propellers. In 1937, then Lt. Cdr. W. D. Leggett (retired as chief of the Bureau in 1955) started working with General Motors on controllable pitch propeller development for World War II landing craft. There has been a number of able officers working on controllable pitch propeller development during all these years and the writer has had the privilege and pleasure of working with all of them and I am still working, presently with Cdr. W. H. Garrett in charge of the Propeller, Shafting, and Bearing Branch of the Bureau of Ships. To make a long story brief, there have been many successes and failures and at the present time the U. S. Navy has 120 ships in operation equipped with controllable pitch propellers. The largest U. S. Navy controllable pitch propeller is 9 ft 9 in. in diameter, absorbing 6850 shp at 300 rpm. From all these years of creative struggle in controllable pitch propeller development there has crystallized a design that appears to be a great improvement over anything else in this line.

The Bureau of Ships for a year and a half has been sponsoring tests on a 20,000 shp full-scale blade actuating unit at a manufacturer's plant. This unit is designed to operate a four-bladed $12^1/_2$-ft-diam, 270-rpm propeller.

The hydraulic system consists primarily of a 21 GPM IMO pump and a 50-hp electric motor both located inside the propeller shaft. In this design the servomotor with four-way valve for changing the blade pitch is located in the hub, and the oil from the IMO pump is circulated through the valve and the servomotor. This forms a closed loop hydraulic system without high-pressure overshaft seals. The unit is extremely compact and highly suitable for ships having limited space. Because of the closed system a pressure of 3000 psi is used in the servomotor, thereby allowing the use of a simpler and more compact hub assembly.

This unit has been operated continuously (450 hr) at 270 rpm with a simulated full load corresponding to 20,000 hp, with oil pressure set at 3300 psi and subjected to a full pitch reversal every minute.

The hub structure and servomotor mechanism are far simpler than any of this general type previously designed and in existence in the world market today. The hub structure has undergone photoelastic investigation in the laboratory and the stresses have been found to be acceptable.

Additional design studies have been completed as follows:

Propeller diameter, ft	rpm	shp
$12^1/_2$	270	20,000
14	200	20,000
18	140	20,000
22	100	20,000

The stresses and difficulties in the design become less as the diameter increases.

The point of this comment is that there is no reason whatsoever for American manufacturers of propulsion gas turbines to worry about all sorts of complicated reversing devices when manufacturers the world over can offer time-proven controllable pitch propellers from the shelf in power ranges from 100 shp to 25,000 shp.

The shipbuilder, however, should be careful in selecting his controllable pitch propeller design because this device is especially attractive to "inventors" as the rapidly increasing number of patent applications indicates.

A controllable pitch propeller is by no means an easy development. It requires years of careful study and experimentation with a most systematic approach in order to achieve the desired result.

B. J. Terrell[9]

The authors are greatly to be congratulated on this paper which must have entailed an enormous amount of careful investigation. It is essentially a reference work and in their classification of the power plants under different headings they have very sensibly accepted a certain amount of repetition, so that each table of data is complete in itself. Although this is a detail, it makes the paper very much easier to read.

The over-all thermal efficiency of the new 5500 shp machinery for *Auris* is given as 28 per cent, whereas it is stated as 26.3 per cent in the recent ASME Paper No. 58—GTP-12 by Lamb and Birts. The illustrations relating to the Pametrada hydraulic reversing system do not relate to *Auris* but are taken from an experimental rig which we have at Wallsend. Fundamentally, however, it is very much the same as that installed in *Auris*, except that there is no provision for an alternator drive.

In the section on maintenance costs on page 313, the figure for steam turbines is quoted as 0.49 cents/kw hour. I think this should be 0.049 cents/kw hr, as stated later on page 335.

In discussing the possibility of fewer engine room personnel in gas-turbine ships, a reduction of one man per watch is quoted as being equivalent to an improvement in thermal efficiency of about 10 per cent on a 20,000 hp cargo vessel. With the quoted fuel cost of $2.70 per barrel, this works out at about $45,000 per man-year. This seems a little on the high side.

On the question of heavy fuel, the paper contains probably the first published report of attempts to burn a really "bad" fuel in the *John Sergeant*. It is interesting to learn that fuel washing was unsuccessful, despite the optimism expressed after the maiden voyage, and it would be most interesting to have a little more information on this subject.

The marine gas turbine certainly seems to be at its best in small high-speed craft, where weight and space are at a premium and power/weight ratio of great importance. I have often felt that a very good field of application for the gas turbine might be in small fast cross-Channel steamers, many of which are only at sea for less than 1000 hr per annum. Here, I think, the gas turbine would yield the advantages of a considerable reduction in space, allowing more passengers to be carried in a given ship; lower stand-by losses because of the immediate availability of the plant; and very probably lower capital cost, because the short life would allow the use of a simple type of machine. The fuel consumption in this type of vessel is not of the greatest importance, but as a rough guide the consumption of the present-day steam cross-Channel vessel is of the order of 0.75 lb/shp hr, all purposes. A very simple gas turbine of the derated, aero-turboprop type could easily better this, although, of course, it would require a distillate fuel. I should be most interested to hear the authors' opinion on this subject.

Authors' Closure

One of the great problems associated with gas-turbine propulsion plants in sizes above 3500 shp has been that of obtaining astern power. The comments by Mr. Strandell lend assurance that the controllable pitch propeller is now in wide use and can be provided in sizes up to 20,000 shp.

In answer to the discussion of Commander Fowden, the initial cost of marine gas turbines has been drastically reduced during the past five years. This applies to the turbines in the 100–500 up range. There have not been sufficient production quantities

[9] The Parsons and Marine Engineering Turbine Research and Development Association, Pametrada Research Station, Wallsend, Northumberland, England.

of any engines above this range to show any cost trends. Production quantities of all marine engines have been extremely small in comparison to engines in the automotive and aircraft fields. Cost in dollars per horsepower has been cut as much as fifty per cent in some engines. With improved manufacturing techniques and increased production the small gas turbine should be in the same cost bracket as the reciprocating engine. Cost of large engines, in the 1000 hp and above range, will be competitive with steam and diesels.

A paper on the peak power gas turbine for a Cosag plant was presented at the ASME Gas Turbine Power Division Conference in Cincinnati, Ohio, March, 1959.

The maintenance cost for steam turbines as mentioned under maintenance on page 313 should be 0.049 cent/kw-hr, as Mr. Terrell suggests, and not 0.49.

The potential savings in operating costs by reduction of one man per watch is considered reasonable when the total costs of maintaining a man (salary and overhead) are evaluated.

Aircraft-type gas turbines would appear to have potential application in ships, such as cross-channel steamers. Mr. Terrell points out that such ships may be at sea less than 1000 hours each year and that fuel consumption is not of great importance. There are a number of turboprop engines now in commercial aircraft service with overhaul intervals between 1500 and 2100 hours. Such an engine could conceivably have as much as two years of service before being removed for overhaul. This would of course place the engine in the same overhaul need as if it were installed in an aircraft. Since the utilization factor would be relatively low and the reliability factor not necessarily as high as in aircraft, an engine might well get three or more years of service before removal for overhaul.

There are certainly areas where combined plants are attractive, as Mr. Markham suggests. Developments in new propulsion systems may have lessened the activities in Cosag and Codag propulsion endeavors. There are naval plants under construction as mentioned in this paper that incorporate both Cosag and Codag. The West German Navy has under construction six fast escort ships equipped with Codag propulsion plants. Each ship has a total of 38,000 shp. There are two shafts and each has two 3000 bhp MAN V8V 24/30 sixteen-cylinder highly pressure-charged four-stroke engines and one 13,000 hp Brown Boveri Company gas turbine.

INDUSTRIAL AND CENTRAL STATION

B. G. A. SKROTZKI

Associate Editor, Power, McGraw-Hill Publishing Company, New York, N. Y. Mem. ASME

Gas turbines have become established components of modern industry. Industrial and central-station units now total 3,614,000 hp installed, planned, or projected, and a wide variety of equipment is available. One generation of commercial industrial gas turbines—the Houdry process—has already become obsolete. Capacity limits of power-generating units are being increased to the 50-mw range, and combinations of gas turbines and steam turbines are offered for 150 mw. Pipeline transport of natural gas and crude oil uses the largest number of mechanical-drive gas turbines. Small gas turbines developed largely for military use have potential for use in industry and commerce.

GAS TURBINES have become established, proved components of modern industry and are no longer experimental units. Fig. 1 shows the remarkable acceptance that gas turbines have won in five short years. Dividing gas turbines into three commercial classes: (*a*) Houdry, (*b*) industrial, and (*c*) central station, we find the following number of units and capacity installed, planned, and projected in the given years:

Table 1 Five-year world-wide gas-turbine growth

	1952		1957	
Houdry units	40	1,277,000 cfm	7	380,000 cfm
Industrial units	66	320,000 hp	349	2,068,000 hp
Central station units	44	430,000 hp	128	1,546,000 hp
Totals	150	750,000 hp	484	3,614,000 hp

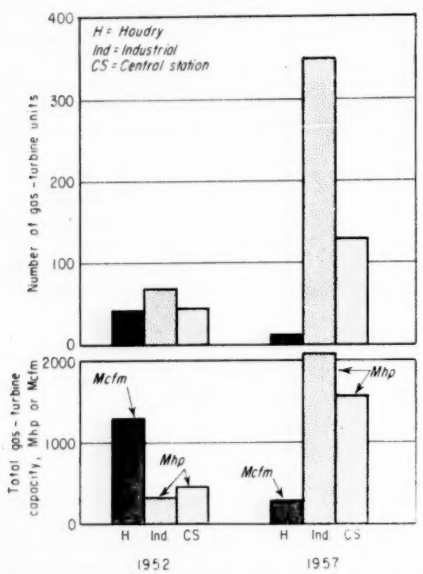

Fig. 1 Comparison of number and capacity of gas turbines installed and projected on world-wide basis for years 1952 and 1957

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, July 1, 1958. Paper No. 58—A-46L.

Houdry Units. These are separated from industrial turbines in this analysis to indicate the "maturity" of the gas-turbine concept. The first unit was installed in 1937, and later ones installed until about 1945 brought the total to 40 units. Most of these served through 1952. Since then advanced cracking methods have made the old Houdry process obsolete, and today only seven are still in service. Thus the first generation of commercial industrial gas turbines have served their purpose profitably and have been retired by advancing technology.

Fig. 1 and Table 1 include gas turbines installed, planned, and projected in the year listed; some of the units listed for 1952 were planned for installation in 1954 and some listed for 1957 were projected for 1960.

Fig. 2 gives the number of units and total capacity installed, planned, and projected in each class of service for *industrial application* of gas-turbine units, as of the year 1957. Electric-power generation and natural-gas pipeline pumping have an equal number of units, but pumping has about one third more total-horsepower capacity.

Oil-field repressuring uses about half as many units as applied to gas-line pumping or to power generation. But repressuring capacity equals about 85 per cent of electric-generation capacity. Oil-line pumping uses only a few less units than repressuring, but has only one third the total capacity.

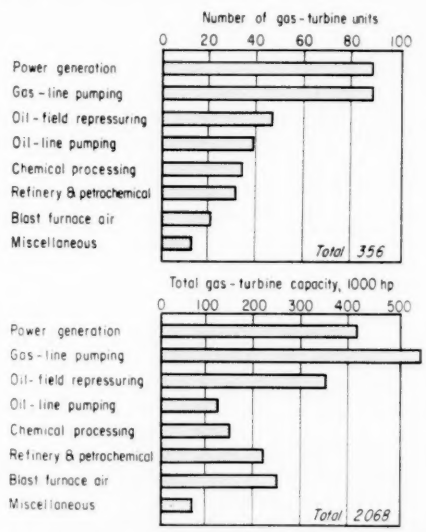
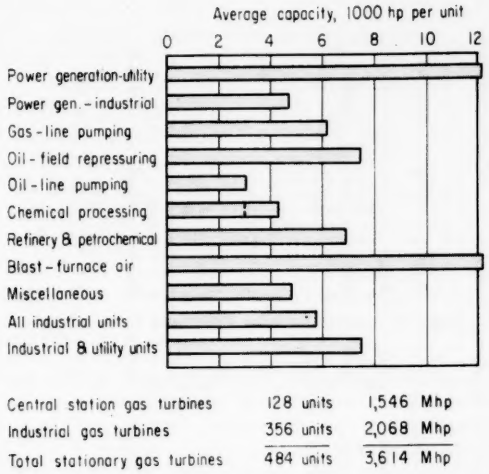


Fig. 2 Planned and existing industrial gas turbines on world-wide basis for 1957 analyzed by types of application



Central station gas turbines	128 units	1,546 Mhp
Industrial gas turbines	356 units	2,068 Mhp
Total stationary gas turbines	484 units	3,614 Mhp

Fig. 3 Average capacity of gas turbines planned and installed as of 1957 analyzed by type of application

Chemical processing uses a few less units than oil-line pumping; but its capacity becomes more difficult to define because the gas turbine as a unit sometimes loses its identity to become a part of the over-all chemical cycle. More about this later.

Refinery and petrochemical units almost equal the chemical processing and the former has a total greater capacity—but again, the same difficulty in capacity definition.

Blast-furnace air supply has the smallest number of applications, but the fourth largest total capacity. Miscellaneous units should be distributed mostly among the classes we just discussed, but inadequate description barred classifying them accurately.

Fig. 3 gives us an idea of the size of gas-turbine units used in each of the industrial-application areas, showing first the average size of central-station units (utility)—12,000 hp or 9000 kw. For comparison, the largest unit projected—40,000 kw—will be built for the Swedish Power Board by Svenska Turbinsfabrik Aktiebolaget Ljungstrom, Sweden, for operation in 1959. Units for blast-furnace air supply have the same average size as the utility units.

Industrial-power-generation units are only about one third as large as the utility units—reflecting the lower individual demands on industrial-electric-power systems.

Oil-line pumping units use the lowest average capacity of the classes listed. Other classes range in between as shown in Fig. 3.

Statistics in Figs. 1, 2, and 3 should only be taken as rough indexes, indicating a minimum value, because figures as received from builders terminated as of different dates with some orders under negotiation that were not revealed.

In addition to the 484 commercial units—industrial and utility—listed in Fig. 3 there are 36 shop and experimental units in builders' facilities with a total capacity of about 107,000 hp.

Available Gas-Turbine Units

The market demand for equipment is reflected in the offering of "standard" models by their builders. This development has been strongly emphasized during the past five years in the gas-turbine field. Table 2 lists characteristic data on some current gas-turbine offerings.

Careful engineering can adapt these models or their components to the needs of a wide variety of industrial processes. In this way the gas turbine becomes economically competitive with other prime movers. Wider fields of application broaden the opportunity for repetitive manufacture and so help reduce the unit cost.

Table 2 does not pretend to be an exhaustive list, but shows the variety of equipment available today. Smaller gas turbines, 500 hp and under, have been applied extensively to military aircraft and naval needs. While they offer possibilities for industrial applications, none have been specifically reported in the survey.

Power-Generating Units

Central-Station Applications. Central-station applications of the gas turbine predicted prior to the 1952 Gas Turbine Progress Report have come to fruition. Units now serve peak-load, end-of-transmission-line, base-load, stand-by, feedwater-heating, and combined gas-steam-cycle duties. Over half the units in the U.S.A. burn natural gas, most of the remainder burn a treated Bunker C; some plants wash and centrifuge the oil to reduce the corrosion-causing salts.

To avoid both turbine-stage corrosion and the use of austenitic materials, most European builders prefer to limit turbine-inlet temperatures to 1200 F and forego the efficiency advantage of higher heat levels. Corrosion difficulties in the 20-mw semiclosed-cycle unit built by Sulzer Brothers at Weinfelden, Switzerland, for the North East Power Company, have led to the dismantling of the plant. While the plant performed well with light oil, corrosion from heavy-oil ash could not be prevented at the 20:1 pressure ratio of the 1200-F semiclosed cycle. Two 10-mw open-cycle single-shaft units with regenerators will replace the 20-mw unit.

Reliability. Gas turbine reliability has been amply proven in service. In one study by H. D. McLean, "Operating experience of General Electric Turbines," ASME Paper No. 58—GTP-18, a group of gas turbines with use factors over 80 per cent were running about 86 per cent of the time, available 98.1 per cent of the time, and had forced outages only 0.7 per cent of the time.

Life of such gas-turbine parts as combustion liners, fuel nozzles, 1st-stage turbine nozzles, 2nd-stage turbine nozzles, turbine buckets and turbine wheels depends on operating conditions. Under favorable conditions of natural-gas firing and continuous loading these parts have run over 30,000 hr with only minor repairs to 1st-stage nozzles and turbine wheels. Under severe conditions of residual-oil firing and intermittent loading, the life of combustion liners is from 3000 to 5000 hr; fuel nozzles over 10,000 hr; 1st-stage nozzles, 5000 to 10,000 hr; 2nd-stage nozzles, turbine buckets, and turbine wheels over 20,000 hr.

Capacity Limitations. Capacity limitations that often bar use of gas turbines on large utility systems are slowly being lifted by advanced designs. Largest unit projected is a 40-mw open-cycle gas turbine to be built by Svenska Turbinsfabrik Aktiebolaget Ljungstrom, Sweden, for the Swedish Power Board. Brown Boveri can now build a 50-mw open-cycle unit in the same space needed for a 30-mw unit. They foresee the likelihood of building 100-mw units in this space using a combined steam-gas cycle with the steam at supercritical pressures.

Of the 109 Brown Boveri turbines, those burning residual oils have water-injection nozzles. These wash ash accumulations from the turbine blades during turbine shutdowns, every 300 to 400 hr.

Brown Boveri has abandoned the double-flow arrangement for high-capacity units as used in their 27-mw Beznau machine and has achieved higher capacity in a single-flow unit by opening the blade angles. This raises capacity from 13 to 30 mw for a given frame size. Another departure for this firm was the offering of units with turbine-inlet temperatures of 1400 F. These units will burn blast-furnace gas and use ferritic steels and more complex blade and wheel-cooling systems, rather than depend on austenitic materials.

Table 2 Characteristic Data of Stationary Gas-Turbine Power Plants

Builder	General Electric Company					Westinghouse Electric Corporation				Brown Boveri Corporation				Ruston & Hornsby, Ltd.	Clark Bros. Co.
Capacity, kw, except where otherwise noted	5000	10,000	16,500	21,800	27,500	2400	3750	8500	16,500	2500	6200	20,000	27,000	650	6200
Turbine temperature, F	1450	1450	1450	1450	1450	1350	1350	1350	1350	1200	1200	1200	1100	1340	1350
Pressure ratio	6.0	6.0	6.0	6.0	6.0	4.9	4.1	6.0	6.0	4.6	4.6	16.0	8.0	4.1	4.25
Cycle	Open	Open	Open	Open	Open	Open	Open	Open	Open	Open	Open	Open	Open	Open	Open
Number of shafts	1	1	1	1	2	1	2	1	1	1	1	2	2	2	2
Shaft speeds, 1000 rpm	6.9	4.9	3.6	3.29		8.5	6.0	4.69	3.6	5.35	3.6	4.75/3.0	3.0/3.0	11.5/6.0	5.0/3.6
Number of turbine cylinders	1	1	1	1		1	1	1	1	1	1	2	3	1	1
Number of turbine wheels in series	...	...	...	...	2	...	2	...	...	...	...	...	DFTC	2	2
Number of compressor cylinders	1	1	1	1		1	1	1	1	1	1	2	4	1	1
Number of intercoolers	0	0	0	0	1	0	0	0	0	0	0	1	2	0	0
Number of heating stages	1	1	1	1		1	1	1	1	1	1	2	2	1	1
Combustor type	Multiple	Multiple	Multiple	Multiple		Multiple	Multiple	Multiple	Single	Single	Single	Single	Single	Single	Single
Regenerator effectiveness, per cent	0	0	0	0		0	75	0	0	75	0	0	80	75	0
Fuel	G, O	G	G, O	G, O	G, O	G, O	G, O	G, O	G, O	BG	G	O	O	O	O
Turbine cooling	Disk	Disk	Disk	Disk		...	...	...	...	D, S	D, S	D, S	...	D, S	D
Compressor type, number of stages	A, 15	A	A, 15	A, 15		A, 14	A, 11	A, 14	A, 15	A	A	A	A	A, 13	A, 13

Builder	English Electric Company, Ltd.	Metropolitan-Vickers Electrical Company Ltd.		Escher Wyss Ltd.	John Brown & Co.	AiResearch Manufacturing Company	Boeing Airplane Company	Solar Aircraft Company			Sulzer Brothers	David Budworth Limited	Utica Division Curtiss-Wright Corporation	
Capacity, kw, except where otherwise noted	1910	1750	2500	2000	2000	100 hp	240 hp	50 hp	520 hp	7500	45 hp	60 hp	2800 hp	
Turbine temperature, F	1430	1180	1292	1220	1220	1550	1500	1100	1460	1256	1480	...	...	
Pressure ratio	4.8	5.0	5.38	3.6	3.6	3.3	3.95	2.35	4.6	6.6	2.7	2.9	...	
Cycle	Open	Open	Open	Closed	Closed	Open	Open	Open	Open	Open	Open	Open	Open	
Number of shafts	2	1	1	1	1	1	2	1	1	1	1	1	2	
Shaft speeds, 1000 rpm	8.25/7.0	7.0	7.0	12.75	12.75	6.0	35.5/2.92	38.25	20.0	4.25	45.0	46.0	...	
Number of turbine cylinders	1	1	1	1	1	1	1	1	1	1	1	1	1	
Number of turbine wheels in series	2	...	...	...	...	...	2	...	...	...	...	...	2	
Number of compressor cylinders	1	1	1	1	1	...	1	1	1	2	1	1	1	
Number of intercoolers	0	0	0	1	1	0	0	0	0	1	0	0	0	
Number of heating stages	1	1	1	1	1	1	1	1	1	1	1	1	1	
Combustor type	Double	Multiple	Multiple	External	Single	Single	Double	Single	Single	Single	Single	Single	Multiple	
Regenerator effectiveness, per cent	60	0	0	90	87	0	0	0	0	77	0	0	0	
Fuel	O	O	O	PC	Peat	DO	O	O, K	O, K	BG	O	O	O, G	
Turbine cooling	D	...	...	...	None	None	...	...	...	...	...	...	...	
Compressor type, number of stages	A-6, C	A, 14	A, 15	C, 3	C, 3	C, 2	C, 1	C, 1	A, 10	A, 21	C, 1	C, 1	...	

A = axial flow
BG = blast-furnace gas
C = centrifugal
D = disk cooling
DFTC = double-flow l-p turbine and compressor
DO = diesel oil
G = natural gas
K = kerosene
O = oil
PC = pulverized coal
S = stator cooling

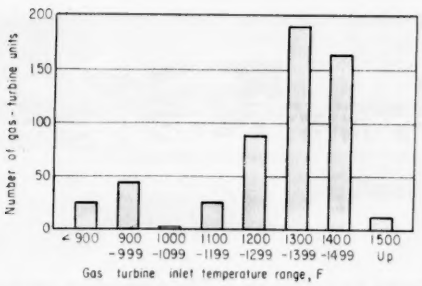


Fig. 4 Distribution of gas-turbine inlet temperatures of existing and projected stationary gas turbines

Three 6200-kw simple-cycle gas turbines, each mounted on two railway cars, were built for the Mexican Federal Power Commission by Brown Boveri. In each unit, one car carries the gas-turbine plant and the second car houses the transformer, tap changer, starting equipment, battery, diesel engine, and small auxiliaries.

Free-Piston Gas-Turbine Plants. Free-piston gas-turbine plants in all types of service use over a total of 90 gasifier units and have accumulated over 250,000 service hours. The largest plant running uses 8 free-piston gasifiers built by SIGMA of France to supply a 6-mw gas turbine, built by Brown Boveri; this is the Cherbourg Plant of the Electricité de France. At full load the gas generators deliver 830-F gas at 43.5 psig to the turbine. Over-all plant efficiency at about 50 to 60 per cent loading averages 34.0 per cent. Largest free-piston plant projected will have six of the Cherbourg units for a total of 36 mw to be installed at Singapore.

Gas-Turbine Developments. Metropolitan-Vickers Electrical Co., Ltd., of England reports that their 2500-kw unit for the Ashford Waterworks, Fig. 5, exceeded guaranteed performance by a comfortable margin. This stand-by unit with 1292-F inlet temperature was guaranteed for a heat rate of 20,400 Btu per kwhr (based on LHV) and an exhaust temperature 795 F. On test the unit heat rate was 19,345 Btu per kwhr and exhaust temperature was only 712 F. This firm has 5 units installed in England, South America, and Borneo with a total capacity of 25,750 kw.

The English Electric Company, Ltd., has built the largest units of all the British manufacturers. These are two 20-mw electric-generating sets at the Royal Aircraft Establishment, near Bedford, England. These supplement the utility supply to drive wind-tunnel compressors. The units employ the basic design features of the 2750-hp plant, Fig. 6, offered by English Electric, namely, axial-centrifugal compressor, 1430-F inlet gas, and two

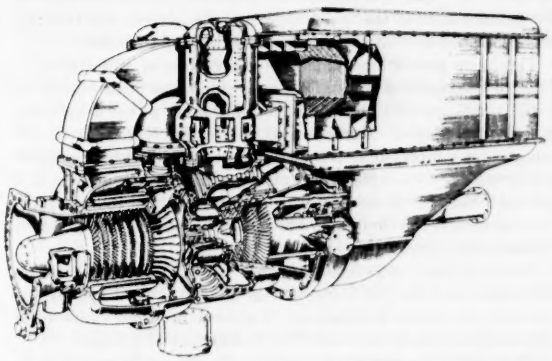


Fig. 6 English Electric Company, Ltd., 2750-hp gas turbine uses axial-centrifugal compressor, two combustors in parallel, 4-stage turbine and regenerator

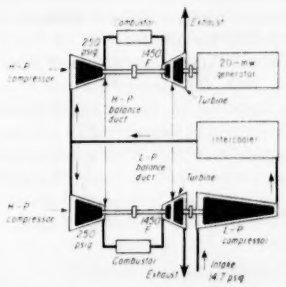


Fig. 7 English Electric Company, Ltd., 20,000-kw gas turbine has unique 4-shaft arrangement with h-p compressors and two combustors in parallel. One l-p turbine drives l-p compressor while the other drives the generator.

large combustors. Unique feature is the 2-shaft parallel arrangement, Fig. 7. The l-p compressor driven by the power turbine of one set discharges to an intercooler, the air then divides equally between the two parallel h-p compressors each feeding its own single combustor. Each combustor feeds a 2-stage h-p turbine driving its h-p compressor. Each h-p turbine exhausts to a 6-stage l-p power turbine, one driving the l-p compressor and the other the 20-mw a-c generator.

The single reverse-flow combustors mount on top of each of the units as in the basic-unit design, Fig. 6, which used two combustors in parallel. Total pressure ratio of the 20-mw unit runs

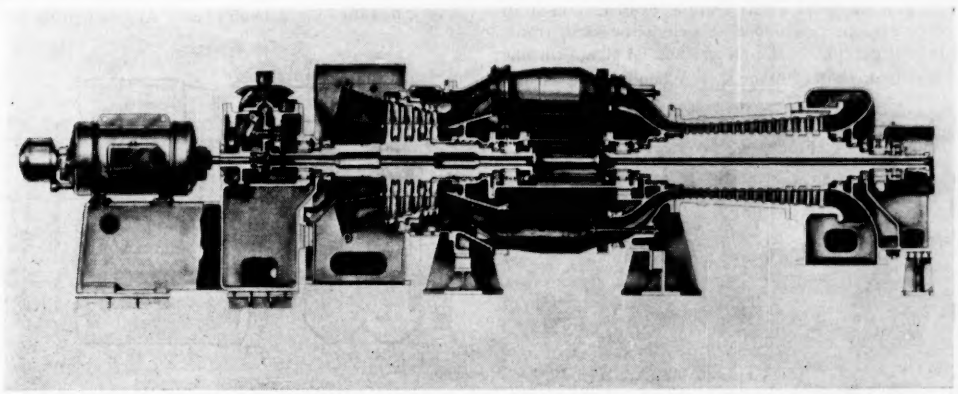


Fig. 5 Metropolitan-Vickers Electric Company, 2500-kw gas turbine uses axial-flow compressor, multiple combustors, and 4-stage turbine

about 17:1. The two parallel components connect through an h-p balance duct at the h-p compressor discharges, and through an l-p balance duct at the discharges of the h-p turbines.

The basic gas-turbine unit offered by Ruston and Hornsby, Ltd., of England has enjoyed a world-wide range of installations, with over 37 installed or ordered for stationary power generation in industrial plants. More than 30 work as mechanical drives for pumps and compressors. These units are offered with or without regenerators; the simple-cycle unit is rated at 785 kw with a 15.2 per cent thermal efficiency and 1340-F turbine-inlet temperature; the regenerative-cycle unit produces 650 kw with a 20.8 per cent thermal efficiency.

Fuels for these units include natural gas, distillate oils, and refinery gas, and one unit burns sewage gas. About 24 use turbine exhaust for steam generation. Eighteen of these units have accumulated a total of more than 66,000 operating hours.

Clark Bros. Company who market the Ruston-Hornsby units in the U. S. A. have a larger unit of their own with a nominal capacity of 8500 hp that they offer for power generation and industrial applications, Fig. 8. Of the 13 units running or on order, three generate electric power; two are rail-mounted emergency power plants—one 5500-kw unit for the U. S. Navy and one 6200-kw unit for the Federal Power Commission of Mexico.

As of February, 1958, the General Electric Company had 160 gas-turbine units installed or on order. Of these 49 generate electric power, 31 of them in electric utility systems. One 16,500-kw simple-cycle single-shaft unit is barge-mounted and serves the Creole Petroleum Corporation on Lake Maracaibo, Venezuela. General Electric is supplying four units for one of the world's largest gas-turbine stations on Vancouver Island, Canada, for the British Columbia Power Commission. The 76-mw station will have two 20-mw simple-cycle single-shaft units and two 18-mw regenerative-cycle single-shaft units. The units will burn treated Bunker-C oil. This plant is expected to be base loaded during its first two years and then work as a peak-load plant with a yearly plant factor of about 25 per cent.

The first combined gas-turbine steam-turbine unit (built by GE) went into service on the system of the Crisp County Commission in Georgia this year. A 5-mw simple-cycle single-shaft gas turbine, burning natural gas, exhausts into a steam boiler generating 600-psig 825-F steam for a 12.5-mw steam turbine. Pulverized coal fires the boiler which can develop full power output for the steam turbine when the gas turbine is not running.

Of some 75 units supplied by Westinghouse Electric Corporation, 19 generate electric power, as of late 1957. Five of these serve electric utility systems, one is a mobile rail-mounted 5-mw unit and the others serve industrial plants.

Studies of the 4700-kw Westinghouse unit installed in Fort Stockton, Texas, of the West Texas Utilities System showed an installed cost of $185 per kw including switchgear yard, transformer, one mile of gas line, land, and grading. A steam-turbine plant would cost more than $200 per kw. When burning natural gas costing less than 25¢ per million Btu, the gas turbine appears to be economically superior to a steam-turbine plant. In the first five years of operation of this unit forced-outage time totaled to only 73.6 hr.

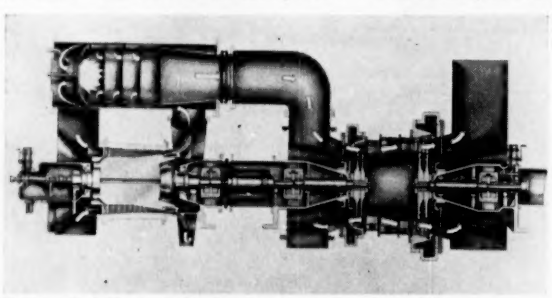


Fig. 8 Clark Brothers 8500-hp gas turbine features axial-flow compressor feeding single combustor, 2-stage h-p turbine driving compressor and 2-stage l-p turbine driving the load

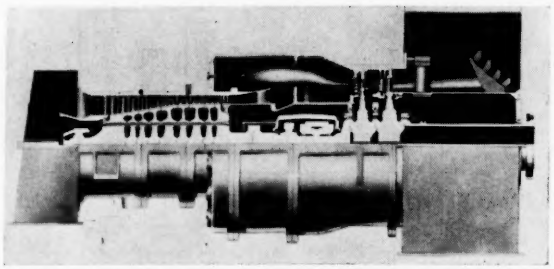


Fig. 9 General Electric Company 21,800-kw gas-turbine plant has single shaft through which 2-stage turbine drives axial-flow compressor, features multiple reverse-flow combustors in parallel

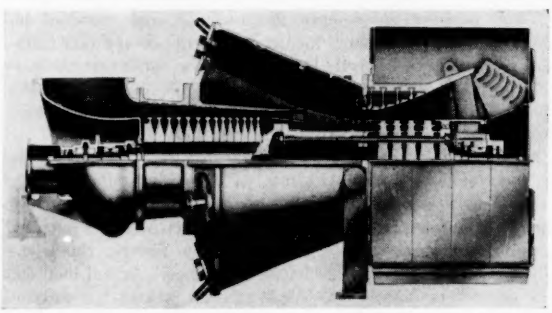


Fig. 10 Westinghouse Electric Corporation 3000-hp gas-turbine plant has 4-stage turbine driving axial-flow compressor, uses multiple reverse-flow combustors

Combined Cycles. Several manufacturers and consulting engineers have made intensive studies of combining gas turbines and steam turbines through supercharged boilers. One Westinghouse study shows that using a 1350-F 16.5-mw gas turbine to supercharge a boiler producing 1800-psig 1000/1000-F steam for a 150-mw steam turbine would improve over-all plant efficiency by about 6 per cent. In addition, the capacity of the steam turbine would be increased by 5 mw because of lower steam extraction for feedwater heating and the gas-turbine capacity would be increased by 5.9 mw because of more gas flowing through the turbine than for a straight gas-turbine cycle. This raises over-all plant capacity of 165.5 mw to 177.4 mw with the same equipment. Investment cost of the combined plant would run about 6 per cent less than for a steam plant. At this writing no plans for

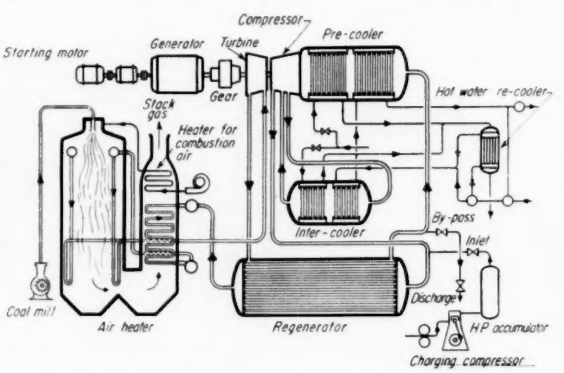


Fig. 11 Escher Wyss Ltd., 2000-kw closed-cycle plant uses pressurized air as working medium heated in pulverized-coal-fired air heater

installing this type of stationary plant have been announced.

Closed-Cycle Gas Turbines. Closed-cycle gas turbines as built by Escher Wyss, Ltd., of Switzerland and by John Brown & Co., Ltd., Scotland under EW license have continued their development. A 2-mw unit at the Ravensburg Plant of EW in Germany has run on pulverized coal since December, 1956, Fig. 11. This air-turbine cycle also supplies space heating for the plant via the cooling water for the precooler and intercooler. The cycle can be restarted in 30 minutes after a week-end shutdown. A 2-mw natural-gas-fired plant went into service in Toyotomi, Japan, in 1957.

Closed-cycle plants under construction or testing are a 12-mw lignite-fired unit for Russia, 6.5 and 13.5-mw lignite-fired units for Germany, and a 2-mw peat-burning unit for Scotland.

Auxiliaries and Controls. Auxiliaries and controls for gas turbines have undergone intensive development, with remote control being a practical reality. Ducting for compressor air inlets has been engineered to minimize pressure-loss penalties in lost efficiency and capacity. Inlet-air filters and silencers used on most units add to the problem of finding a balance between acceptable pressure loss and lower compressor maintenance.

Discharge ducting must also keep down pressure loss at the turbine exhaust to maintain turbine efficiency and capacity. Stack selection and placement must pay special attention to possible recirculation of exhaust to compressor inlet. Ducting design employs turning vanes, expansion joints, supports, acoustical insulation, access manholes, and heat insulation to meet the needs of a smooth-running plant.

Both dry and viscous-type filters are used on air inlets for replacing dirty with clean areas. These can be automatic or fixed. Standard silencers are available for smaller gas turbines—but larger sizes usually need individual engineering of this unit. Building design, duct supports, type of cycle all bear upon silencer effectiveness.

Cranking power for gas turbines may be supplied through a variety of equipment depending on the location. These include electric motor, steam turbine, gas-expander turbine, or internal-combustion engine.

Where available, water may be used to cool turbine parts, lubricating oil, or generator. Where water is scarce the turbine and its auxiliaries may be completely air cooled.

In hot climates, evaporative coolers lower inlet-air temperature to better the gas-turbine efficiency and capacity. Investment in cooler equipment usually proves profitable for the capacity gained.

Gas turbines may be engineered to burn only gas or oil or both alternately. For the dual-fuel arrangement, the fuel can be switched while the turbine is under load. Turbines burning residual oils usually start on a distillate oil because of easier ignition and to prevent residual oil solidifying in the lines during shutdown. When preparing the turbine for shutting down, distillate oil flushes out the fuel system before the operator extinguishes the combustor flame.

Industrial Gas Turbines

Gas-turbine versatility shows itself in the wide range of industrial applications, Fig. 2. The gas turbine first served in this area in pressurizing Velox boiler furnaces and as part of the Houdry oil-refining process; since then new applications have

Fig. 12 Ten 5300-hp General Electric gas turbines drive natural-gas compressors in this plant at Lake Maracaibo, Venezuela, owned by the Creole Petroleum Corporation

been developed. The majority of installations produce shaft power for generating electricity or driving mechanical equipment. The newer applications provide a mechanical shaft-power link in a process that salvages heat energy from one part to activate another part. The shaft power developed may be more, equal to, or less than the energy needed to activate a process. Accordingly: (*a*) The excess shaft power is a byproduct, or (*b*) the process is ideally balanced, or (*c*) additional shaft power must be supplied by another prime mover.

Mechanical Drive. Pipeline transport of natural gas and crude oil uses the largest number of mechanical-drive gas turbines, Fig. 2. In May, 1949, Westinghouse Electric Corporation placed an experimental 1800-hp gas turbine in service driving an Ingersoll-Rand direct-connected centrifugal gas compressor at Wilmar, Ark., for the Mississippi River Fuel Corporation. After serving this pipeline for over a year the unit was shifted to Farmington, Mo., and in January, 1951, began "packing" an underground-storage field. This unit has accumulated over 50,000 service hours to establish a world's record.

Largest concentrations of gas turbines are the repressurizing plants for the Creole Petroleum Corporation in Lake Maracaibo, Venezuela. Fig. 12 shows the first plant consisting of ten 5300-hp 2-shaft simple-cycle gas turbines built by General Electric Company. Recently completed was a plant with twelve 7000-hp units; these are single-shaft simple-cycle units. A plant with fourteen 7000-hp units is under construction to start running in 1959. The latter two plants use Westinghouse Electric Corporation gas turbines. The plants pressurize natural gas to almost 2000 psig for reinjection into the oil-bearing strata.

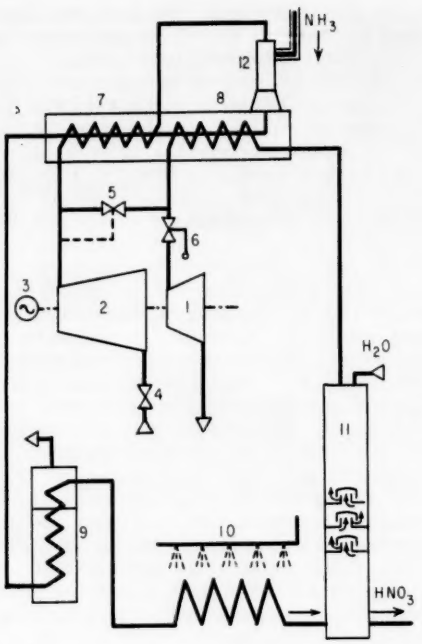
Process Units. While most of the original Houdry process units have been retired, other methods of oil cracking still use high-pressure air for catalyst regeneration. About 12 units are now in this service; the circuits are similar to the old Houdry process described in the 1952 Gas Turbine Progress Report. These generally produce an excess of shaft power used for electric generation.

Variations in using gas turbines appear in processes for making nitric acid and butadiene. Some 64 units will work in refinery, chemical, and petrochemical plants. Brown Boveri has been very active in the nitric-acid-plant applications as described by Z. Stanley Stys in "Gas Turbines for the Chemical Industry," ASME Paper No. 57—GTP-9. Fig. 13 shows the layout of a process cycle built by the Chemical and Industrial Corporation of Cincinnati, Ohio, for the Mississippi Chemical Corporation at Yazoo City, Miss. An 8-stage expander (gas-turbine) drives a 9-stage centrifugal, intercooled isotherm compressor that delivers 18,000 scfm of air at 120 psig. Fig. 14 shows the unique isotherm compressor which has seven stages of intercooling. An axial compressor is not used in this cycle because of the high-pressure ratio. Intercooling reduces the shaft-power demand of the compressor.

In Fig. 13 the compressed air rises to 550 F in the preheater before mixing with vaporized ammonia over a platinum-gauze catalyst. The columns and towers separate the nitric oxide and nitrogen, and the nitrogen passes through the expander or gas turbine, at about 75 psig after being heated to about 900 F, to develop shaft power for driving the compressor. In this cycle, the turbine cannot develop enough power to drive the compressor so an auxiliary motor must make up the deficiency.

In older plants using reciprocating compressors, an input of about 350 kwhr per ton of acid produced was needed. The gas-turbine cycle of Fig. 13 needs an input of only 155 kwhr per ton and this can be brought down to 126 kwhr per ton. The auxiliary motor could be eliminated by using a larger compressor and paralleling a combustor with the bypass valve, number 5 in the illustration, so the turbine could develop all the power needed by the compressor.

Blast-Furnace Compressors. Blast-furnace compressors driven



1= Exit-gas expansion turbine
2= Isotherm compressor
3= Auxiliary geared motor
4= Throttle
5= Bypass valve
6= Quick-acting valve
7 = Air heater
8 = Exit-gas heater
9 = Waste-heat boiler
10= Cooler-condenser
11= Absorption tower
12= NH_3 converter

Fig. 13 Nitric-acid-plant cycle has gas-expansion turbine 1 supply part of shaft work needed to drive intercooled compressor 2, auxiliary motor 3 supplies remainder of power input needed by compressor

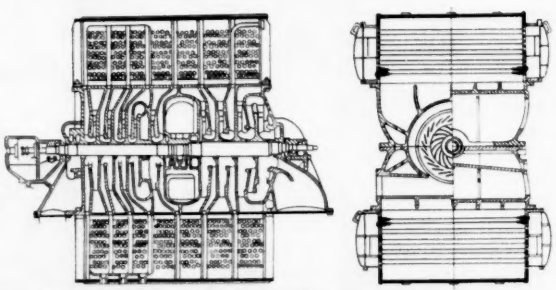


Fig. 14 Brown Boveri 9-stage intercooled centrifugal compressor (isotherm compressor) uses seven stages of intercooling. Air enters at right to pass through five stages toward center, then flows to outer left and exhausts from 9th stage, at middle of compressor, to enter nitric-acid cycle.

by gas turbines have given good service. About 24 units now work or will soon be employed in this area, producing pressurized air for blast furnaces or converters exclusively, or producing enough shaft power to also drive an electric generator. Flexibility also marks the gas turbine in this application which allows variation in over-all arrangements. These units burn blast-furnace gas with a heating value of about 80 to 100 Btu per cu ft. They also use a stand-by fuel, usually distillate oil.

Fig. 15 shows a 125,000-cfm unit being installed by Westinghouse Electric Corporation for the U. S. Steel Corporation. The 1350-F gas turbine drives two axial-flow compressors. One compressor pressurizes blast-furnace gas from 8 in. water up to 52

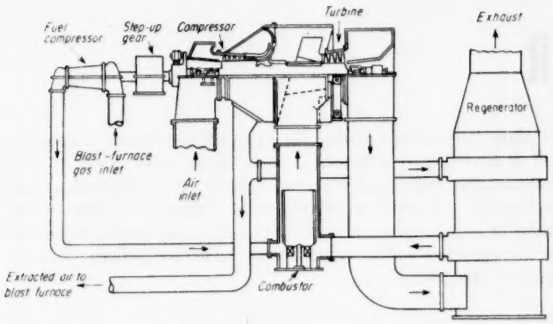
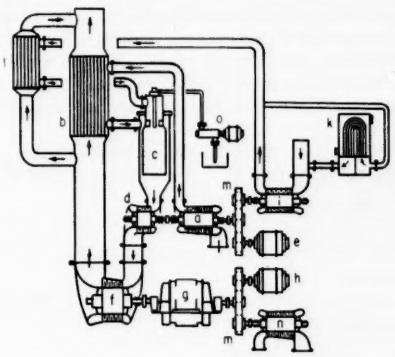


Fig. 15 Westinghouse single-shaft gas turbine uses common axial-flow compressor for turbine plant and for supplying 125,000-cfm air to blast furnace



(a) Axial air compressor
(b) Air preheater
(c) Combustion chamber
(d) Compressor turbine
(e) Starting motor
(f) Power turbine
(g) Generator
(h) Exciter
(i) Blast-furnace gas compressor
(k) Blast-furnace gas cooler
(l) Blast furnace gas preheater
(m) Gearing
(n) Blast-air compressor
(o) Fuel-oil pump

Fig. 16 Brown Boveri gas-turbine cycle to produce 5000-kw electrical output and 30,000 cfm of blast-furnace air—2-shaft layout gives good part-load efficiency

psia and the other raises air from atmospheric to 48.5 psia. Air from the main compressor is divided, with 38 per cent to the cold-blast main for the furnaces and the balance going to the regenerator for heating from 360 to 840 F. Turbine exhaust through the regenerator cools from 950 to 540 F.

Fig. 16 shows a dual-service unit that produces a net electrical output of 5000 kw and 30,000 cfm of blast-furnace air at 35 psig. This unit was built by Brown Boveri Corporation for Niederrheinische Hütte, Duisburg, Germany. The combustor burning blast-furnace gas discharges at 1400 F into the h-p turbine which drives the gas-turbine compressor and the blast-furnace-gas compressor. H-p turbine exhaust enters the l-p turbine which drives the electric generator and the blast-furnace-air compressor.

The l-p turbine exhausts into the paralleled regenerator and fuel-gas preheater. Pressure ratios are: 5.0 for the gas-turbine compressor, and 2.7 for the blast-air compressor. The 2-shaft arrangement gives good part-load performance.

Those units that have been running for some time are all outside of the U. S. and have given good service.

Combined Industrial Cycles. Combined industrial cycles have also been applied or planned in the past few years. These use single-shaft gas turbines to develop mechanical power and discharge the turbine exhaust into a fuel-fired boiler to generate steam for steam turbines.

The Gates Rubber Company in Denver, Colo., has a General Electric 5-mw gas turbine that exhausts into a 120,000-lb-per-hr boiler. At 4400-mw load the turbine exhausts 310,000 lb per hr of 828-F gas to the boiler furnace. This waste heat can generate 35,000 lb per hr of 200-psig 600-F steam. The boiler can run at 200 to 300 psig to produce process steam or at 450 psig to supply steam turbines. Supplementary natural-gas firing produces the full capacity of the boiler. A forced-draft fan feeds air to the boiler furnace when the gas turbine is out of service.

Union Carbide Chemicals Company Division of Union Carbide Corporation, Texas City, Tex., will use a 7000-hp gas turbine to drive a multistage centrifugal air compressor arranged in two cases with two stages of intercooling and an aftercooler. These will supply 100 and 200-psig air for plant use. Turbine exhaust will supply air to a pressurized-furnace steam boiler to generate about 300,000 lb per hr of 1000-psig 850-F steam. The turbine combustor will burn pressurized by-product gas with a HHV of about 710 Btu per cu ft. Supplementary fuel for the boiler will be natural gas. The steam drives a 9375-kw back-pressure steam turbine that exhausts at 200 psig and 450 F.

Small Gas Turbines. Small gas turbines, some of which are listed in Table 2 have been developed largely for military uses. They range from 5 hp to 500 hp and burn distillate oils. While some industrial applications of the smaller units have been mentioned, no details have been released.

Specific military-service applications suggest where they might be used in civilian industry and commerce. These include: Emergency-generator drives, auxiliary-generator drives, pump drives, deicers, pressurized-air supply, and marine, vehicle, and aircraft drives. Compared to small reciprocating internal-combustion engines, gas turbines suffer in fuel economy. But, where compactness, light weight, and high speed supersede the need for fuel economy, the small gas turbine has much to offer. This particularly applies to stand-by-service installations.

The smallest capacities invariably use radial-inflow turbines, while the larger ones feature axial turbines with 2-shaft arrangements for better part-load efficiency. These units have no more than two combustors in parallel; the smaller capacities have only a single combustor and some feature an annular combustor. Most of the units have single-stage centrifugal compressors, but some of the larger have 10-stage axial-flow compressors.

There appear to be over a thousand of the small-size gas turbines in service, practically all of which are military.

Nuclear Energy. Up to this writing no nuclear-powered stationary gas-turbine plants have been built. Louis H. Roddis, Jr., formerly of the Atomic Energy Commission has reported that in 1954 the Aircraft Reactor Experiment, a liquid-salt-fueled and cooled reactor was operating at Oak Ridge, Tenn., at temperature levels that could have run a gas turbine. Actually the heat was dissipated to atmosphere. In 1956, an aircraft gas turbine was running on nuclear energy at the National Reactor Testing Station in Idaho.

One AEC contract involves the building of a high-temperature gas-cooled power reactor which could presumably be used for a gas turbine. In addition three design study contracts have been let by the AEC for closed-cycle gas-turbine-propulsion plants with gas-cooled reactors.

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NUCLEAR

The nuclear gas-turbine plant has recently become the basis for a substantial development program. Because of considerable difficulties, particularly in the areas of heat and radiation-resistant materials, from 5 to 7 years must elapse before any real accomplishment can be demonstrated. The most attractive line of development from a cost, efficiency, and weight point of view, appears to be the single-fluid gas system, discussed in detail. The case for the optimum gas is not clear-cut. Thermodynamic, system, and bearing and seal problems are considered.

NUCLEAR FISSION as a source of mechanical power has advanced in the past 10 years from the status of physical possibility to that of large-scale accomplishment. The rapid progress thus far is most encouraging in light of the widespread and growing deficiency of natural and fossil-fuel sources.

All nuclear industrial-power systems known to be in service today employ a steam cycle as the proximate means of generating useful output. The first practical types of power reactor, including the pressurized-water-moderated type, imposed a rather low upper limit on working-fluid temperature, in the neighborhood of 600 F. A saturated-steam system could be provided to operate usefully within this limitation at about 24 per cent thermal efficiency. This type of cycle was employed in most of the "first generation" nuclear power plants in various countries of the world. More recent advances, notably in boiling-water reactors and in high-temperature gas-cooled reactors coupled to steam boilers, enable the steam-cycle conditions to be raised toward a more efficient level. It may be assumed that in time the steam conditions associated with nuclear power will more nearly approach the standards which now apply to fossil-fuel plants.

The compatibility of the gas-cooled reactor with the closed-cycle gas turbine is readily recognized. The "stack-loss" problem, which plagues the fossil-fuel-fired closed cycle, disappears entirely. A single fluid serves as coolant and power medium, providing a rational and basically efficient power cycle. Both reactor and turbine are today limited to a similar maximum operating temperature, about 1300 to 1500 F. Should improved materials permit this limit to be raised, outstanding performance will become possible.

Technical Considerations in Nuclear Gas-Turbine Systems

Cycles. A reactor as a heat source is capable of serving an open cycle as well as a closed one. Use of an open cycle has not been attractive for several reasons. First, bulk and cost of reactor core and shielding cannot be justified except by intensive generation of heat and power, and an open system is limited in this respect. Also, radioactive contamination of the atmosphere would be unavoidable and most difficult to reduce to the point where it would be acceptable.

Two-fluid systems, employing a primary coolant to deliver heat to a secondary gas-turbine plant, have received some attention. The turbine cycle then may be open or closed, but the systems investigated to date have not proved highly attractive. The primary coolant may be a liquid metal or a gas, but there is a parasitic loss of power in circulating this coolant. Also, two added heat-transfer processes are introduced. The result is a reduction of turbine-inlet temperature, to such a low level that a desirable efficiency cannot be obtained. Weight is also greatly increased when separate fluid systems are used. This is a serious disadvantage for some installations such as ship-propulsion machinery.

Table 1 [9][1] presents a comparison of the following cycles on the basis of total weight for a plant developing 20,000 shp:

1 Direct closed cycle with CO_2.
2 Direct closed cycle with helium.
3 Indirect closed cycle with CO_2 as reactor coolant and air as power-plant fluid.
4 Indirect closed cycle with CO_2 as reactor coolant and CO_2 power-plant fluid.

Table 1 Weight of propulsion plants for different cycles, in lb

Cycle	CO_2	Helium	CO_2–Air	CO_2–CO_2
Propulsion-machinery	300,250	305,880	304,600	302,300
Reactor plant	914,070	907,050	1,331,490	1,187,350
Shielding[a]	1,133,500	974,500	945,500	945,500
Electric plant	176,500	176,500	310,000	310,000
Independent-system machinery	85,900	85,900	85,900	85,900
TOTAL	2,610,220	2,449,830	2,977,490	2,831,050
Weight to power ratio, lb per shp	131	122	149	141

[a] These shielding weights are based on the use of liquids for secondary shielding. For each of the cycles listed, a weight credit of 520,000 lb has been taken for diesel fuel oil used also as shielding.

The most attractive line of development from a cost, efficiency, and weight point of view appears to be the single-fluid gas system, and some principal problems of such plants are discussed herein.

Selection of Working Fluid. The fact that the gas system is to be closed and pressurized allows the selection of the most advantageous working fluid—not necessarily air as in open-cycle turbines. However, this opportunity so far has not resulted in a unanimous preference for any one fluid. The merits of various gases are qualified and sometimes contradictory, and cannot be expressed by a single figure. Since any choice is a compromise, the best selection depends upon the application.

Gases most frequently proposed include air, nitrogen, carbon dioxide, and helium. These coolants will be discussed briefly in relation to various factors which are significant in a large power application such as ship propulsion.

Contributed by the Gas Turbine Power Division and presented at the Annual Meeting, New York, N. Y., November 30–December 5, 1958, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society. Manuscript received at ASME Headquarters, August 22, 1958. Paper No. 58—A-46M.

[1] Numbers in brackets designate References at end of paper.

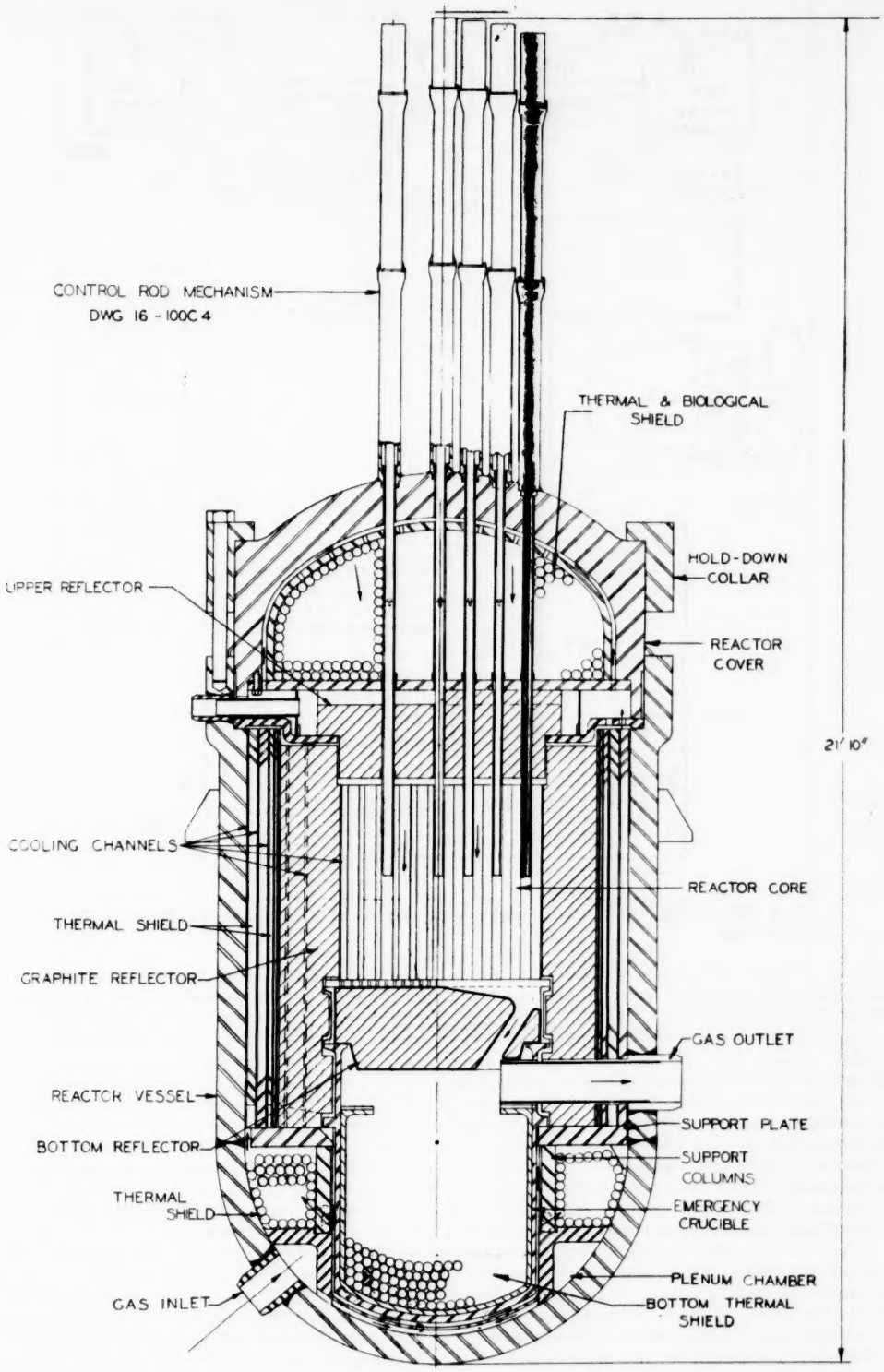

Fig. 1 A typical reactor using carbon-dioxide cooling gas, located in the cycle as shown in Fig. 2, over-all height 21 ft, diam 7 ft; designed for heat rate of 55 mw. The core is made up of a lattice containing about 330 fuel assemblies with graphite moderator; space is provided for coolant-gas flow, up the outside and down through the core.

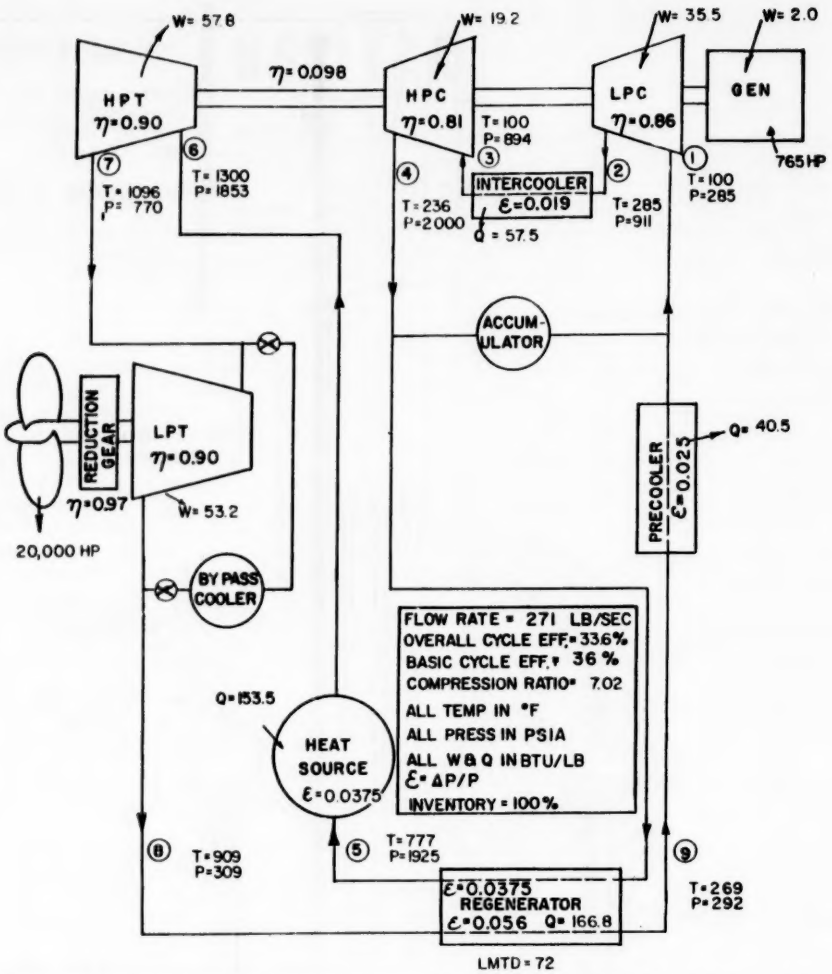

Fig. 2 Carbon-dioxide closed-cycle gas-turbine power plant of 20,000 hp showing heat balance at 100 per cent load

Availability. The absence of a supply problem might dictate the selection of air for an installation, either military or civilian, where extreme isolation was one of the conditions. In such cases it may be expedient to overlook other shortcomings of air as a working fluid.

Both nitrogen and CO_2 are commercial products, easily available, and present no problem in this regard. A nitrogen system could in an emergency accept air, if necessary, without detriment to its mechanical operation. The same is true of a CO_2 system, although a limit on output would be imposed.

Only in the case of helium is there complete dependence upon a narrow channel of supply. The sources of helium are restricted geographically and in this country are under government control. Also, a system designed for helium will accept no other fluid even in an emergency. Any program which is predicated on using helium requires firm assurance that availability will continue, both for the prototype and for subsequent construction. There is, in addition, the problem of logistics: The supply must be made effective at the time and place where it is needed, which could be on board ship at some remote port. An ample reserve is vital in these circumstances. The difficulty is aggravated by the readiness with which helium escapes through any crevice.

Chemical Activity. The moderator material most readily adaptable for use in a gas-cooled reactor is graphite. A disadvantage of graphite, however, is that, at 800 F or higher, it is subject to oxidation in the presence of air, with the formation of CO and CO_2. The situation with respect to CO_2 is similar, a part of the gas being reduced to CO in the hot region of the reactor. On passing into a cooler region, a part of the carbon is deposited, the effect being a gradual transport of solid material out of the reactor into heat exchangers or other apparatus where lower temperatures prevail. This condition can be corrected by canning, cladding, or coating the graphite. Fortunately cladding appears quite feasible. Some of the better-known heat-resistant alloys, which depend on their content of Ni and Cr, appear to be adequately resistant to oxidation at 1500 F and perhaps higher, and offer good prospects for cladding purposes. Other moderators have been considered, for example zirconium hydride, but indications are that cladding will also be necessary in such cases. Some added enrichment of the nuclear fuel is normally required to compensate for neutron absorption by the cladding material.

Pure nitrogen of course would not lead to graphite oxidation but it is known to cause nitriding of steel at a temperature of 900 F or above, and the gas cannot be considered inert. The

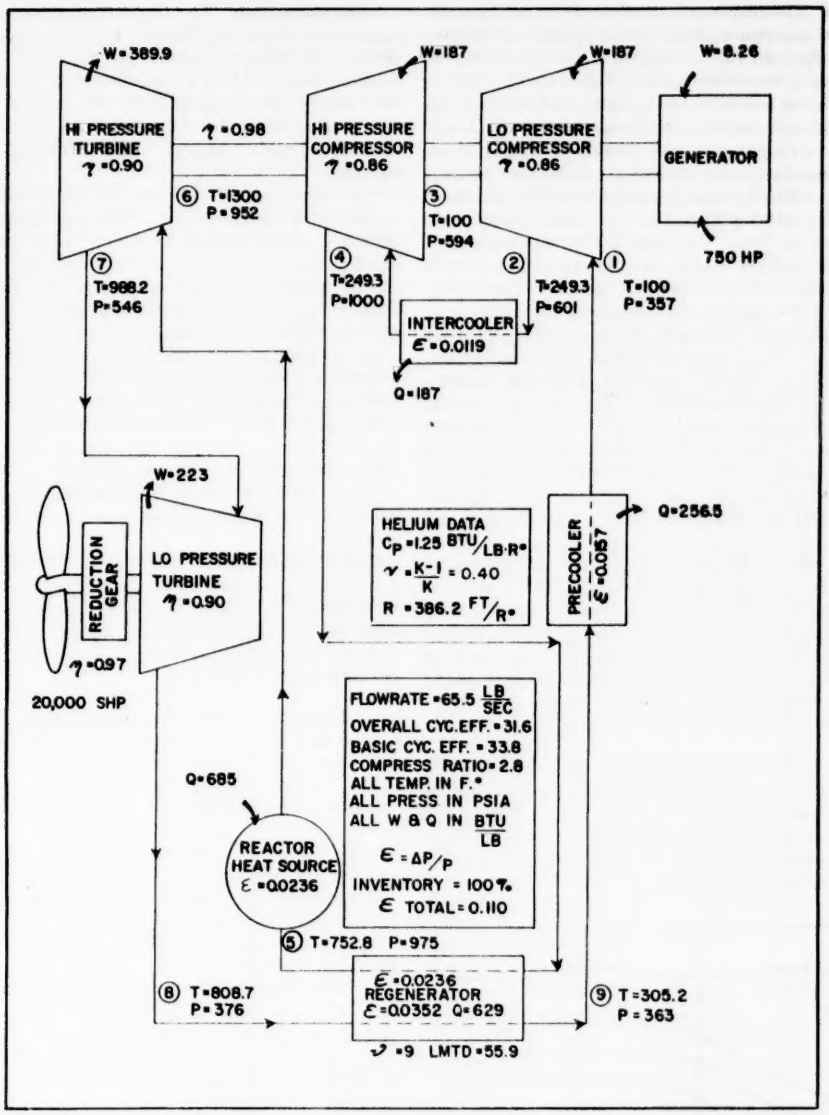

Fig. 3 Helium closed-cycle gas-turbine power plant of 20,000 hp showing heat balance at 100 per cent load

addition of $^1/_2$ per cent of O_2 helps to inhibit this process, but again may introduce a need for cladding the moderator.

Helium offers the advantage of complete inertness, although a few tests [11] have indicated that some heat-resistant materials tend to become embrittled when exposed to pure He at 1600 F. The writers are not aware of an explanation of this peculiar behavior. Possibly the occurrence of oxidation at exposed grain boundaries, which is usual in an air atmosphere, has a protective effect.

Radioactivity. The argon content of air and of commercial N_2 is the potential cause of seriously harmful radioactivity. The isotope A^{41} will be formed, with strong gamma radiation (1.4 Mev) and a half-life of 109 min. Even 10-ppm A, representing a specially purified grade of N_2, has been estimated to require 0.6-in. lead shielding under certain conditions to reduce the gamma radiation to presently accepted limits [9]. In contrast, air has a normal A content of 8500 ppm. Leakage of such radioactive gas into a confined space would rule out human occupancy. Also, the extended half-life of A^{41} would prevent access to machinery for several hours after shutdown.

Carbon dioxide becomes very strongly radioactive because of the formation of N^{16} in a radioactive field, and considerable shielding would be required because of the resulting 7-Mev gamma radiation. However, the health hazard from release of this gas would not be as great since the half-life of N^{16} is only 7 sec. Personnel would be able to enter the shielded compartment very soon after reactor shutdown.

Due to its near zero neutron cross section, helium does not become radioactive and in this sense is ideal. Some impurities, such as traces of argon, are certain to be present and of course must be accounted for. Since rupture of a fuel element could result in scattering of radioactive material through the power system regardless of the gas used, compensating features must be provided in any case.

Thermodynamics. The characteristics of air and of nitrogen are familiar from past experience. Both carbon dioxide and helium depart substantially from the behavior of air, and some of the effects are apparent in the cycle diagrams, Figs. 2 and 3.

Fig. 2 [9] represents a marine power plant based on CO_2 in an unusually high-pressure system—2000-psia maximum. Compression takes place near the saturation region of the gas, where specific volumes may be no more than 80 per cent of their perfect-gas values. The reduced power demand for compression is reflected in the "work ratio" of the cycle, with output turbine work being a fraction of the total. The ratio is 48.0 per cent for the CO_2 cycle, against 36.3 per cent for the He cycle. Where perfect gas characteristics prevail, see Fig. 3. The CO_2 cycle is more efficient and less sensitive to component deficiencies.

The helium cycle has its optimum efficiency at a low pressure ratio of about 2.4, a consequence of the high specific-heat ratio, $k = 1.67$. A large number of compressor stages are required, about 40 in all, for an optimum cycle. This may be a rather large item of cost, but is not expected to cause unusual technical difficulty. Volume changes in He machines are small and it should be simple to avoid stall. Blade-tip speeds are limited by mechanical considerations alone, since the velocity of sound in He is about 3400 fps and Mach ratios will not generally exceed 0.35.

Helium is an efficient heat-transfer medium, being excelled only by hydrogen. This is a substantial asset, for the heat-exchanger apparatus of a helium system is smaller by perhaps 30 per cent than that of a CO_2 system. The advantage is pronounced at temperatures below 600 F. However, CO_2 holds a clear advantage when cooling a very compact reactor core due to its special heat-transport ability. Nitrogen and air are inferior heat-transfer fluids and offer no such strong compensating advantage.

In order to show the excellent efficiency of the closed-cycle system, note in Fig. 2 that at full load the CO_2 system has an over-all cycle efficiency of 33.6 per cent while in Fig. 3 helium has 31.6 per cent. For part-load operation, which is very necessary in many nuclear-power applications, the efficiency of both systems remains high as shown in Figs. 4 and 5.

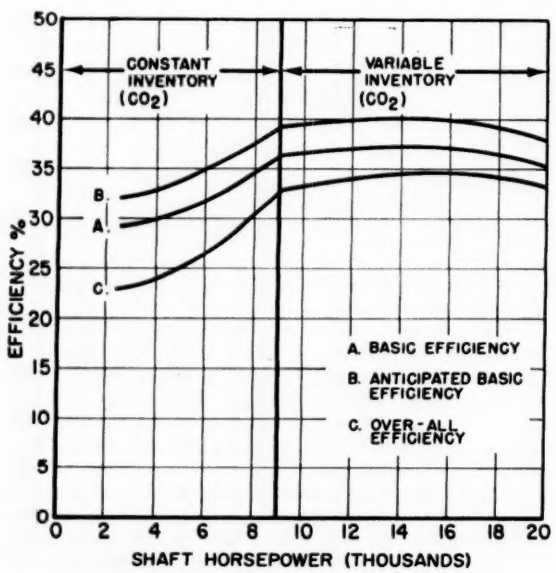

Fig. 4 Efficiency of closed-cycle gas-turbine power plant using carbon-dioxide gas

System Components. The combination of high pressure and high temperature favors the "barrel" type of compressor and turbine design. In this construction the outer pressure housing is a simple cylinder without a horizontal split. Compressor and turbine stators, split horizontally in the usual way, are contained within these housings. This construction makes it possible to eliminate external leakage. Minor internal leakage can be tolerated.

A desirable method of making gas connections to the reactor is by way of a concentric duct. The high-temperature stream is carried in the inside pipe, the wall of which is subject only to a small pressure difference. It is expected that similar principles can be applied in the high-pressure turbine and in the required shut-off valves. The objective is to insulate the hot-gas stream from the high-pressure containment walls.

Heat exchangers of simple tubular counterflow design are promising. This type provides minimum cross section, an important consideration in high-pressure containment.

Bearings and Seals. Efficient seals at shaft-penetration points are mandatory. Also, contamination of the working fluid with oil or oil vapor must be held to a minimum. While these objectives are difficult to attain, it appears that sealing devices can be developed or existing systems refined to meet the new requirements.

Gas bearings are attractive in principle but their development has not yet fitted them for use in heavy machinery. Efforts are continuing in this direction and it is hoped that a dry, rotating-shaft seal for high-pressure gas service will become available. The oil-contamination problem, however, will probably persist in the foreseeable future.

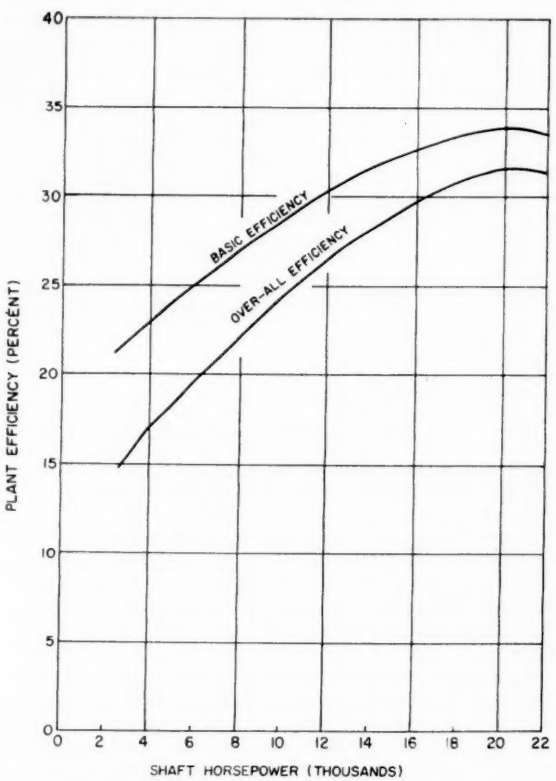

Fig. 5 Efficiency of closed-cycle gas-turbine power plant using helium gas

Summary

Based on present technology, selection of the optimum working fluid for a direct-cycle, gas-cooled, reactor and gas turbine can be made only after weighing the advantages and disadvantages of the several conflicting characteristics. There seems to be no single fluid which is the obvious best choice for all applications, but each must be considered on its own merits for individual cases.

The gases covered in this paper—He, CO_2, N, and air—represent the known fluids which can be utilized in a practical manner. Of these He is the only one which presents a problem regarding availability; the others are obtainable from commercial sources.

Regarding chemical activity, He is the best choice since CO_2 and air will cause oxidation in the presence of graphite, and N has undesirable nitriding effect. Regarding radioactivity, He is also the best choice, being a perfect material in this respect. The efficiency of a CO_2 cycle is high, one reason being the reduced power required for compression. Helium has better heat-transfer characteristics, but leakage would be more of a problem with this fluid.

As can be seen, the case is not clear-cut in the selection of the optimum gas, and a great deal of work will be necessary to properly resolve this question.

Tabulated Data and Comments From Specific Study Reports

Within the past several years, there has been a growing interest in the combination of nuclear power and gas-turbine systems. The first known related proposal was made at Oak Ridge in 1944 by Farrington Daniels. Work on this proposal was suspended in 1947. Most of the effort in developing nuclear-power systems was expended on other systems, notably pressurized-water types. However, improved technology and the undeniably high potential of gas turbines prompted a revival of interest in gas cycles for power generation, and now a number of programs are under way to exploit this undeveloped but promising means of providing power.

There are two major types of nuclear gas-cooled reactor systems under development at the present time: The first, typified by Calder Hall in England, is the indirect-cycle type where a reactor-coolant gas passes through a heat exchanger, forming steam for operation of conventional machinery; the other, which is primarily covered in the following studies, consists of a direct-cycle system wherein reactor-coolant gas is utilized directly to operate gas-turbine machinery. Of the two, the second type certainly offers the better promise for the future for gas-turbine machinery and reactor design require only the metallurgical break-through in high-temperature materials to realize their ultimate capabilities and efficiencies. This high state of efficiency will be reached at operating temperatures well above those experienced today.

Eight preliminary studies which have been prepared in recent years are listed in Tables 2 and 3. It should be emphasized that these are studies only, most of them quite preliminary in nature, and are not representative of nuclear gas-turbine plants either under construction or in operation.

Study No. 1, 1947, For: Atomic Energy Commission
Designer: F. Daniels, at Oak Ridge National Laboratories
Application: Power production and breeder for plutonium production
Type of cycle: Indirect—helium to steam
Plant efficiency: 20 per cent
Reference: Mon N-383, "Summary Report on Design and Development of a High Temperature, Gas-Cooled Power Pile" [1].

This system, although not strictly a gas-turbine cycle, is of interest in that it represents a pioneer gas-cooled-reactor project as well as one of the first serious attempts in a program that has become known as "Atoms for Peace"; that is, power production from nuclear energy. This system was first proposed at Oak Ridge in 1944 and was the subject of considerable design effort in 1945–1946. Work was then discontinued and no further effort devoted to design and construction of this power pile.

Study No. 2, December, 1954, For: American Turbine Corporation (company supported)
Designer: American Turbine Company (Escher-Wyss)
Application: Electric-power generation
Type of cycle: Direct—closed-cycle
Plant efficiency: 40.4 per cent
Reference: Design study of "60-mw Closed-Cycle Gas-Turbine Nuclear Power Plant" [2].

The design description for this plant gives no data for the reactor other than heat rate, pressure, and temperature, and the conclusions must be appraised in this light.

The compressor and turbine units described in the report were originally designed for air. The entire machinery set, h-p—l-p compressors, h-p turbines, and independent l-p turbines, are shown contained in a single casing, so that the whole set, including precooler, intercooler, and recuperator, measures only 25 by 23 ft by 17-ft 6-in. high in maximum dimensions. A starting motor and electrical generator would be external to these dimensions.

Control is normally accomplished by varying system inventory at different power levels. A low-pressure turbine-bypass valve is available for transients or emergencies, and a cool-down loop with circulator for removing decay during shutdown.

Although the report stresses the reliability and conservative design of these gas turbines, no mention is made of the serious bearing and seal problems associated with helium.

Study No. 3, March, 1956, For: Private study
Designer: F. Daniels
Application: Power production and breeding U^{233} from thorium
Type of cycle: Indirect—helium to air
Reference: *Nucleonics*, vol. 14, March, 1956, pp. 33–39 [3].

In this proposal, Dr. Daniels stated that gas-turbine technology is far enough along to permit utilization of all-gas plants, but proposed a heat exchanger in the first one for safety reasons. The 5-mw (electrical) output was selected because such a unit was ready for production when the article was written and would have required less development.

For the future, Dr. Daniels proposed a number of possible developments, among which was a relatively small (10-mw) plant for nonindustrialized areas. When some operating and experimental data have been obtained, this plant could be converted to a direct-cycle nitrogen plant. In long-range planning, Dr. Daniels recommended research toward development of a molybdenum or tungsten turbine capable of withstanding temperatures of 3000 to 4000 F, which could be used with a uranium-carbide-fueled, helium-cooled reactor to exploit the superior thermodynamic properties and efficiencies of high-temperature machinery.

Study No. 4, February, 1957, For: U. S. Army Engineers
Designer: Sanderson & Porter
Application: Full-scale test loop for portable nuclear-power plant; conditions given for the reactor and turbine are for the ultimate power plant where available
Type of cycle: Direct—closed-cycle
Plant efficiency: 16 per cent minimum
Reference: Design report, "Closed-Cycle Gas-Turbine Test Facility" [4].

The machinery set for this plant is basically an Escher-Wyss design with the turbine-compressor set operating at 18,000 rpm. A reduction gear reduces this to 1200 rpm for driving the 480-volt, 60-cycle 400-kw, 0.8-power factor a-c generator. It is estimated that the ultimate (reactor-powered) plant will have a gas-transit time of 0.957 sec.

Table 2 Reactor data

Studies	1	2	3	4	5	6	7	8
Designer of plant	Daniels	Am. Turbine	Daniels	S. & P.	Ford Inst.	Ford Inst.	G.M.C.	G.D.C.
Coolant	He	He	He	Nitrogen	Nitrogen	He	He	CO_2
Thermal rating	12 mw	148.5 mw	20 mw[a]	—	44 mw	64 mw	55 mw	43.7 mw[b]
Type	Thermal	—	Thermal	—	Thermal Heterogeneous	Thermal	Thermal	Thermal
Inlet temp	500 F	886 F	820 F	755 F	760 F	781 F	768 F	777 F
Outlet temp	1400 F	1400 F	1350 F	1150 F	1300 F	1150 F	1300 F	1300 F
System pressure	150 psi	1000 psi	225 psi	210 psi	535 psi	596 psi	1000 psi	2000 psi
Fuel enrichment	30% UC_2		UC_2	—	10% UO_2	4.4% UO_2	100% UO_2	29% UO_2
Moderator	BeO		Graphite	H_2O	Graphite	Graphite	Graphite	Zr Hydride
Reflector	Graphite		Graphite		Graphite	Graphite	Graphite	Graphite
Shield			Concrete 60,000 cu ft		Concrete	Fe—H_2O Laminations [c, f]	H_2O, oil & Lead [d, f, g]	I—Fe & H_2O II—oil & H_2O [d, h]
Control		[c, d]		[c, e]				

[a] Flow 103,000 lb/hr. [b] 55 mw maximum. [c] Constant pressure (Variable inventory). [d] L-p turbine bypass.
[e] Turbine-reactor-recuperator bypass, compressor bypass for shutdown.
[f] Variable-pitch propeller.
[g] Thermal control and throttle valve. [h] Up to 40% power variable reactor outlet temperature. 40–110% power constant reactor outlet temperature-variable inventory.

Table 3 Data of gas-turbine unit

Studies	1	2	3	4	5	6	7	8
Designer of plant	Daniels	Am. Turbine	Daniels	S. & P.	Ford Inst.	Ford. Inst.	G.M.C.	G.D.C.
Working fluid	Steam	He	Air	Nitrogen	Nitrogen	He	He	CO_2
Plant output	2400 kw	60,000 kw	5000 kw	400 kw	15,000 kw	22,000 shp	22,000 shp	20,000 shp
Flow rate, lb per hr	—	780,000	470,000	66,600	1,100,000	474,000	295,000	975,000
Pressure, max.	450 psi	1015 psia	420 psi	210 psi	530 psi	600 psi	1000 psi	2000 psi
H-p Inlet								
Temp	735 F	1400 F	1250 F	1150 F	1300 F	1150 F	1300 F	1300 F
Pres	450 psi	985 psia	400 psi	203 psia	510 psi	578 psi	962 psi	1853 psi
L-p Inlet								
Temp		1160 F	—	—	—	—	997 F	1096 F
Pres		665 psia	—	—	—	—	552 psi	770 psi
Compressors								
No		2	1	2	1	1	3	2
Intercooling		Yes	Yes	No	Yes	No	Yes	Yes
Inlet								
Temp		90 F	72 F	100 F	90 F	90 F	100 F	100 F
Pres		426 psia	110 psi	76 psia	125 psi	300 psi	362 psi	285 psi
Outlet								
Temp		210 F	244 F	329 F	235 F	295 F	196 F	236 F
Pres		1015 psia	420 psi	210 psia	535 psi	600 psi	1000 psi	2000 psi
Pres. ratio		2.382	3.82	2.745	4.3	2.0	2.76	7.02
Heat exchangers								
Precooler		Δt 176 F	293 to 72 F	Δt 319 F	Δt 205 F	Δt 259 F	Δt 182 F	Δt 169 F
Intercooler		210 to 90 F		—	Δt 175 F		I Δt 96 F II Δt 96 F	Δt 185 F
Recuperator		Δt 676 F	244 to 743 F	Δt 425 F	Hot Δt 525 F Cold Δt 525 F	Δt 486 F	h-p Δt 527 F l-p Δt 550 F	h-p Δt 541 F l-p Δt 640 F

The set is normally controlled by variation in gas inventory through a receiver-compressor-accumulator set. Transients can be accommodated by opening a bypass across the reactor-turbine combination which decreases heat impact but retains full compressor load. On shutdown, a bypass and circulator across the compressors maintains a minimal flow through the reactor to prevent damage.

Investigations of gas-lubricated bearings are under way in order to eliminate oil problems which must be overcome for satisfactory operation. Nitrogen purity for this plant is specified as follows:

Allowable impurities:		
	Argon	50 ppm
	O_2	20 ppm
	Water vapor	1 ppm
	All others	10 ppm

Study No. 5, April, 1957, For: Atomic Energy Commission
Designer: Ford Instrument Company
Application: Power generation
Type of cycle: Direct—closed-cycle
Plant efficiency: 34 per cent
Reference: Final report, "15-mw Gas-Cooled Closed-Cycle Reactor-Power System Study" [5].

This report covers a study on the feasibility of a 15-mw gas-cooled reactor plant for power generation. A considerable effort was devoted to theory and core-physics design, but no extensive work was undertaken in the design of machinery. Many calculations and physical constants are given; however, considerable experimental and computer work would be necessary to advance the program beyond the stage outlined in this report.

A significant outcome of the study was that although Ford Instrument originally selected nitrogen as their coolant, information gathered subsequent to that decision indicates that a re-evaluation is in order, and helium probably will be substituted [6].

Study No. 6, April, 1957, For: Atomic Energy Commission
Designer: Ford Instrument Company
Application: Propulsion for 20,000-shp nuclear tanker

Type of cycle: Direct—closed-cycle
Plant efficiency: 25.7 per cent
Reference: Ford Instrument Report No. 102, Final Report—"20,000-Shp Nuclear Tanker" [6, 7].

In this study the Ford Instrument Company concentrated on reactor design and general plant considerations, while the machinery was discussed in a portion of the report prepared by the Nordberg Manufacturing Company. Nordberg was not enthusiastic about the selection of helium as a working fluid since it would require larger machinery and an extensive design and development program. Their choice would be nitrogen or carbon dioxide.

Nordberg also devoted a good deal of attention to bearings and seals, recommending the use of a double-bushing (liquid-film) type for this plant. The oil film under pressure would provide the gas seal, and a back-up face seal would provide protection on shutdown or bushing-seal failure. Gas bearings were also discussed, but Nordberg did not recommend them on machinery of this size due to the precision required in clearances, finish, and alignment, and the almost instantaneous failure characteristic of the unit. Piping for the plant was described as double-wall construction as developed by Escher-Wyss to keep the outer carbon-steel wall at about 150 to 175 F and minimize thermal stresses.

Study No. 7, April, 1957, For: Atomic Energy Commission
Designer: General Motors Corporation
Application: Tanker propulsion
Type of cycle: Direct—closed-cycle
Plant efficiency: 29.8 per cent
Reference: Final Report, "Study Contract for 20,000-shp Nuclear-Propulsion System for Tanker" [8].

As in several other gas-cycle nuclear plants proposed recently, GMC selected helium for a coolant gas, primarily due to its inertness with regard to chemical and nuclear reactions. Over-all, the report is exceptionally complete and includes a wealth of design detail on items such as turbines, compressors, blading, seals, and the reactor itself. No serious problems are foreseen in the design of helium machinery, and it was considered that the seal problem, although complex, should be adequately resolved by a thorough development program.

Control of the plant would be primarily automatic, being accomplished basically by variations in the reactor-outlet temperature. In addition, an l-p turbine-bypass and throttle valve were included in the loop to provide control of transients. Ship-speed variations and reversal are accomplished with a controllable-pitch propeller.

Study No. 8, April, 1957, For: Atomic Energy Commission
Designer: General Atomic Division, General Dynamic Corporation
Application: Ship propulsion
Type of cycle: Direct—closed-cycle
Plant efficiency: 34.5 per cent
Reference: Feasibility Report, "Gas-Cooled Reactor for Ship Propulsion" [9].

In this study General Dynamics recommended the use of CO_2 as a coolant, based on its high efficiency as a heat-transport medium at elevated temperatures, its compatibility with reactor materials, and its ready availability in high purities. Use of CO_2 also would help simplify machinery-design and sealing problems, but would result in increased secondary shielding.

A simple accumulator system is a feature of the recommended plant-control system, in the higher power ranges. Below 40 per cent full power, reactor output would be regulated by variations in reactor-outlet temperature. When its parameter reaches its maximum value (1300 F), further increase is handled by increase in mass flow by bleeding CO_2 from the accumulator into the compressor suction. A decreasing power demand causes gas to be returned to the accumulator from the h-p compressor discharge. In this way, the compressor is used to pump gas into or out of the accumulator.

The core life was estimated at 465 full-power days. A potential advantage of the CO_2 is that, if all coolant is lost due to some accident, air can be substituted and the plant operated at reduced power (limited by radiation-safety considerations), until a new supply of coolant is available. Such is not the case with helium plants due to their reduced shielding and other design requirements.

Acknowledgment

The authors wish to express their appreciation to the Atomic Energy Commission, and to the General Atomic Division of General Dynamics, for permission to utilize the information available in their files.

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